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REPORT USAAVSCOM-TR-76-27

BEARING RESTORATION BY GRINDING

A. Manufacturer's Viewpoint

B. Evaluation of Ball and Roller Bearings Restored by Grinding

C. Specification for Restoring Bearings by Grinding

D. A User's Viewpoint

E. Microeconomic Analysis of Military Aircraft Bearing Restoration

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Proceedings of Joint Army-NASA Seminar

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A joint program was undertaken by the NASA Lewis Research Center and the Army Aviation Systems Command to restore by grinding those rolling-element bearings which are currently being discarded at aircraft engine and transmission overhaul.

Three bearing types were selected from the UH-1 helicopter engine

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(T-53) and transmission for the pilot program. Groups of each of these bearings were visually and dimensionally inspected for suitability for restoration. A total of 250 bearings were restored by grinding. Of this number, 30 bearings from each type were endurance tested to a TBO of 1600 hours. No bearing failures occurred related to the restoration by grinding process. The two bearings failure which occurred were due to defective rolling elements and were typical of those which may occur in new bearings. The restorable component yield to the three groups was in excess of 90 percent.

The risk and cost of a bearing restoration by grinding programs was analyzed. A microeconomic impact analysis was performed. The program will result in the Government obtaining bearings at lower cost and equivalent reliability.

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PROCEEDINGS OF

JOINT ARMY - NASA SEMINAR

BEARING RESTORATION BY GRINDING

May 20-21, 1976

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ARMY AVIATION SYSTEMS COMMAND

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FOREWORD

The U.S. Army Aviation Systems Command (AVSCOM) is one of several major commodity commands under the newly designated U.S. Army Development and Readiness Command. AVSCOM is responsible for the technical management and supply support of over 9 500 military aircraft, primarily helicopters. These aircraft are assigned not only to active army units, but also include those operated by the National Guard and Reserve components. The AVSCOM mission includes continuous efforts to identify and implement improved methods for the support of the user while reducing the total cost of ownership of all assigned items of equipment.

One potential for extending the service life and reducing the cost of ownership for Army aircraft was suggested to AVSCOM by the NASA Lewis Research Center at Cleveland, Ohio. Their concept was to restore, through grinding, those bearings which, under normal overhaul and maintenance procedures, would be discarded. Their technical analysis of the feasibility of restoring rolling-element bearings through grinding indicated that most such bearings could be returned to a new condition when reworked under a controlled process. They also indicated potential cost savings that could be attained through the application of the bearing restoration process.

A joint Army-NASA task was initiated to identify, locate, and subsequently performance qualify rolling-element bearings restored through grinding under a pilot program. After considering a wide range of candidate bearings for the program, it was decided to select bearings used on the UH-1/T53 airframe/engine system. One bearing selected was the split-inner-ring mainshaft ball bearing from the T-53 engine. A second bearing selected, also from the engine, was the radial roller mainshaft bearing. The third bearing selected was the triplex input pinion ball bearing from the UH-1 main transmission.

A total of 529 of the above three types of candidate bearings, removed from the engines and transmissions during the overhaul process, were identified for the restoration program. All of these bearings were closely examined by the bearing contractor, Industrial Tectonics, Inc. (ITI), prior to being restored in order to record all pertinent visual and dimensional data. In addition, all ring elements were subjected to magnaflux and hardness testing to further substantiate the baseline from which all future events could be judged. From the original quantity of 529 bearings, 250 were introduced into the restoration

process although approximately 93 percent of the initial quantity of bearings was judged by ITI to be restorable. This consisted of 50 bearings each of the two engine bearings and 150 of the triplex bearings discussed above.

An extensive program was undertaken to "requalify" these bearings under a controlled program of laboratory inspections and tests, and by actual engine and transmission test stand operations with the refurbished bearings installed. All qualification testing was conducted independently of ITI. The NASA-Lewis Research Center prepared the technical specification for the bearing rig testing portion of the program. The Corpus Christi Army Depot prepared and conducted the engine and transmission simulated operational tests. The results of this joint program are to be presented at this seminar.

Consideration of the economics involved in the application of bearing restoration will also be presented. An economic analysis has been prepared by the NASA-Lewis Research Center. It is a guide for all interested organizations when deciding on bearings that can be cost effectively restored. The immediate benefits are obviously related to high initial cost or replacement cost of the bearings. The cost of restoration is also a function of the number of bearing replacements being generated by the fielded equipment. That is, the cost of restoration per bearing restored is closely related to the setup of an economical quantity of bearings for the grinding process. Not all bearings would qualify for consideration as candidates on an economics basis. However, there is a cross-over point on many bearings at which it becomes more economical to restore than to replace the bearing.

In summary, the results of this joint program indicate that there are many potential aviation and nonaviation applications for restored bearings within the various military equipment programs. The objective of this seminar is to provide the knowledge gained by AVSCOM to all organizations that may benefit by the implementation of this process into their new and on-going maintenance programs.

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RESTORATION BY GRINDING OF AIRCRAFT BALL AND ROLLER BEARINGS - A MANUFACTURER'S VIEWPOINT

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ABSTRACT

A study was initiated by AVSCOM in conjunction with the NASA and CCAD to undertake a rolling-element bearing restoration by grinding pilot program. A 90 percent yield can be achieved by the restoration process. The estimated cost savings to restore rolling-element bearings ranges from 27 to 47 percent of new bearing cost. Lot quantity for bearing restoration should correspond to the rate at which bearings are removed from operation and become available. Initial start-up time from signing of a purchase order to the first delivery of restored bearings is six months.

INTRODUCTION

Bearing refurbishment has been practiced in various forms by both the military and the airlines for the last 20 years. The rapidly rising cost of engine and transmission bearings in today's more sophisticated turbomachinery has made it a matter of economic necessity to reconstitute aircraft bearings by whatever viable techniques are available.

Bearing refurbishment takes many forms depending on the facility which undertakes it. In most cases bearings are disassembled, cleaned, visually inspected and if no print discrepancies, flaws or major imperfections are found, the bearings are reassembled, lubricated and packaged for further service.

In some cases new rolling elements are inserted. Those bearing refurbishment facilities which have the capability to do so, will also introduce a superficial polishing operation on the raceways to remove minor raceway imperfections and

*Work performed under AVSCOM contract no. DAAJ01-73-C0501 (P3L).

blemishes. However, polishing of raceways, unless done by special machines and under controlled conditions, can destroy raceway geometry, and surface conditions can be created which can adversely affect life. It can therefore be dangerous under some circumstances to perform such a salvage polishing operation on high-performance turbomachinery bearings used in today's jet engines.

Restoration by grinding of rolling-element bearings removed from aircraft engines and transmissions at scheduled overhaul, is a new concept developed in its present form by ITI in conjunction with the NASA Lewis Research Center, the Army Aviation Systems Command and the Corpus Christi Army Depot. The objective of bearing restoration is to reduce the number of bearings which are unnecessarily scrapped at overhaul and thus achieve considerable cost and raw material savings. Bearing restoration represents a logical extension of the practice of bearing refurbishment. The process entails grinding of races and other critical surfaces of used bearings to their original functional characteristics and dimensions. Grinding can remove raceway imperfections to significant depths below the surface. In addition, this stock removal is accomplished by grinding races on the same machines and with the same controls as are used in the manufacture of new bearing raceways. The result is geometrical accuracy and surface finish identical to new bearing raceways.

The restoration by grinding process can be approached in two ways: (1) as a means of reconstituting bearings with imperfections beyond the salvaging capability of the ordinary bearing refurbishment process; (2) as a sole means of reconstituting all used bearings to original functional characteristics and dimensions. The decision as to which approach to use is mainly one of economics, and one of the level of race imperfection which a given aircraft application can tolerate.

Bearing refurbishment as practiced by the airlines and military services has already demonstrated the effectiveness of this salvage operation in reducing bearing replacement costs for aircraft engines and transmissions. Restoration by grinding will demonstrate further savings. The process will prove its economic merits especially in reconstituting the more complex and thus more expensive aircraft bearing required for today and tomorrow's more advanced and sophisticated aircraft engines and transmissions. Furthermore, these advanced engines and transmissions cannot operate reliably with reconstituted bearings which do not possess the original design accuracy which restoration by grinding

achieves.

The objective of this paper is to describe the restoration by grinding process which the ITI Bearing Division has developed, the economic impact, and the manner in which such a program can be implemented and operated. The program was performed by Industrial Tectonics Bearing Division under AVSCOM Contract no. DAAJO1-73-C-0501(P3L).

OVERHAUL PRACTICES

Bearings removed at engine and transmission overhaul are cleaned and visually inspected for defects and dimensional conformance to print. Only those bearings are taken apart and subjected to detailed visual inspection which are by design, separable. Therefore, impending or minute fatigue spalling cannot be detected except by hand feel or noise testing. Both of the latter techniques are very subjective and can only be used with any degree of success by a bearing manufacturer and then only after reference criteria has been established on large production runs of a given bearing.

Table I shows the rejection rate of bearings removed from turbine engines and transmistions of the UH-1 helicopter. This data was taken over a three month period and shows that the engine bearing rejection is 56 percent, and the transmission bearing rejection rate is 47 percent.

One major airline alone sends out 2400 jet engine mainshaft bearings and up to 12 000 accessory bearings per year for refurbishment. This particular airline has in operation at any one time approximately 750 JT8D engines which are used on 727, 737 and DC-9 aircraft primarily. These are larger engines than the type used in the UH-1 helicopter and the bearings used are more complex and more expensive.

A schematic layout of the main body of the engine is shown in figure 1. OEM prices for the bearings in this engine range from $120 for the relatively simple #4-1/2 position roller bearing, to $750 for the #4 split-inner-ring ball bearing duplex set. Since these bearings are furnished exclusively by the engine manufacturer, who warrants them, the price that the airlines pay for these bearings is almost doubled. Refurbishing mainshaft bearings alone represents an annual savings of close to $500 000 and to this figure can be added approximately another $500 000 in savings for the refurbishment of all accessory bearings.

Only 845 mainshaft bearings are replaced annually to keep 750 engines flying. This is a very low figure particularly since commercial utilization and return

on investment per engine requires their almost continuous usage. An equivalent military engine would use up bearings at perhaps three times that rate. This is due to two factors: (1) commercial engines are traditionally over designed and (2) they do not operate under conditions as severe as those required of military engines. A helicopter transmission imposes perhaps the greatest operational loads on its bearings of all aircraft applications.

The reliability of refurbished bearings which have been put back into service in commercial jets has been phenomenal. The experience cited by major airlines indicates that never knowingly had a refurbished bearing failed in engines between scheduled overhaul periods. Similar experience has been reported by the Naval Air Rework Facility in San Diego (ref. 1).

Based upon experience, approximately 10 percent of the bearings which are rejected fail by what is referred to as classical rolling-element fatigue. This failure mode is the basis upon which bearing life is predicated (ref. 2). As a result, approximately 90 percent of the rolling-element bearings which are being rejected for reuse in aircraft application can be refurbished or restored and reused. In other words, of the 4212 UH-1 bearings which found their way into the scrap bin it is probable that approximately 3800 could be salvaged and reused (table I).

Bearing fatigue life has increased considerably in the last 20 years due to the combined effort of the bearing industry, users, and Governmental independent research laboratories (refs. 2 and 3). The improvements are mainly due to increased cleanliness of the steel achieved by more sophisticated steel melting techniques which has greatly reduced impurities and to improved manufacturing processes. As a result, it is anticipated that bearings failing from fatigue will be an even smaller percentage of those bearings rejected at overhaul. This will result in a greater number of bearings which can be refurbished or restored.

RESTORATION BY GRINDING

The ITI process of bearing restoration by grinding must be done on the same equipment and by the same skilled personnel as the manufacture of new turbomachinery ball and roller bearings. The process constitutes approximately the last 30 percent of the total operations required to make a new bearing. The process consists of the following steps:

(1) Visual inspect all components to determine if the bearing is restorable.

(2) Grind and/or chrome plate and grind as required to original print size and tolerances the outer ring o.d. and inner ring i.d. These are the bearing mounting dimensions.

(3) Grind and/or lap, as required, all faces and lands. These may then become undersize by 0.001-inch maximum which is a nonsignificant variation and not functionally important.

(4) Remove by grinding 0.002 inch per side of stock on all race surfaces. Hone surfaces as required to achieve original print surface finish and race accuracy.

(5) On split-inner-ring (gothic arch) bearings only, remove by grinding sufficient stock on the two mating inner faces to restore proper "shim dimension" which restores bearing to correct contact angle, axial and radial play, and resting angle.

(6) On roller bearing races, up to 0.002 inch per side of stock is removed by grinding of the channel faces.

(7) Manufacture new balls 0.004 inch oversize, and new rollers 0.004 inch oversize on the o.d. and width. The identical material specification is used to produce these new rolling elements as in the original bearing.

(8) All silver plating is stripped from the separators. The separators are visually and non-destructively inspected for defects. Ball and roller retention is reworked as required to retain oversized rolling elements. The separators are silver plated and dynamically balanced as required. In the case of two piece separators or deep groove ball bearings, new rivets are used.

(9) The visual and dimensional inspection techniques, procedures and processes, as well as the inspection fixtures and gages used, are identical to those used for the manufacture of new bearings. Similarly, all non-destructive inspections performed are to the same specifications and procedures as those regularly employed in the manufacture of new bearings. This includes, as applicable: nital etch and magnaflux of rings and rolling elements, magnaflux and die check of separators, X-ray and ultrasonic mainly on materials.

(10) Bearings restored by grinding are subject to the same quality control system as are all other rolling-element products produced. The ITI Quality Program, as defined by the ITI Quality Program Manual, is applicable to all contracts received, whether for products or services, and applies to all subcontracts issued under such contracts. The program conforms to Military Specification MIL-Q-9858A

"Quality Program Requirements" and its supplements such as QRC-82G as required by specific customer contracts. Incorporated in the Quality Program are MIL-C-45662A "Calibration System Requirements" for all gages and instruments. All nondestructive testing conducted conforms to MIL-STD-271D and NTR-1A.

(11) All components and the final assembly of the restored bearings are inspected in the same manner as new bearings produced. Bearings are marked to identify the fact that they have been restored by ITI. They are packaged to the original specification and shipped to their destination. Bearing serialization and material traceability as well as inspection records are maintained on permanent file as required. Government and customer source inspection is performed as requested.

Figure 2 shows a disassembled view of a typical jet engine mainshaft bearing of split-inner-ring construction. This is the most complex of all aircraft bearings as far as internal geometry is concerned. Figure 3 shows the surfaces which are restored by the restoration by grinding process. Both inner and outer raceways are reground to a depth of 0.002 inch per surface. Since these race radii are now 0.002 inch larger the bearing must be refitted with new balls 0.004 inch larger in diameter.

The effective race curvature after restoration as defined in figure 3 will be identical to the original dimensions within significant mathematical values. Regrinding the split-inner-ring mating faces, or shim dimension, restores the bearing to its original values of contact angle, resting angle, end and radial clearance. Although the bearing now contains oversized balls and oversized raceway curvatures, the total effective geometry of the bearing has not been changed and consequently the stress level, calculated bearing life and high speed dynamics of the restored bearing will be identical to the original bearing.

Roller bearings are processed in the identical manner. Figure 4 shows a disassembled view of a typical aircraft roller bearing and figure 5 shows the corresponding engineering drawing. Rollers having 0.004 inch oversize diameters are manufacturered in order that a 0.002 inch depth of grinding can be accommodated on both inner and outer raceway. The roller length will also be oversize to accommodate the regrinding of the channel faces.

With radial roller bearings, it is necessary to fit not only oversize diameter rollers into the reground races, but also to assure that an oversize width roller is used such that the print tolerances for channel-to-roller-end clearance is maintained.

Roller bearing manufacturing has the added complexity of roller contour. Figure 6 shows a typical aircraft bearing roller. Almost all high performance aircraft rollers are partially crowned. The new rollers used to restore a bearing are manufactured with the same crown radius, crown length and surface finishes as rollers used for new bearings. Roller manufacturing in the restoration by grinding program is done on identical machines by identical procedures as any new roller for new aircraft roller bearings.

Records of all restored bearings shipped are kept by original part number, restoration number, serial number and date of shipment in order to have these inspection records available at all times.

PILOT PROGRAM

An investigation was conducted of 529 bearings comprising three different types removed at overhaul from UH-1 helicopter engines and transmissions. These bearings under normal circumstances would not be reinstalled for use in the aircraft nor refurbished for later reuse. The purpose of the pilot program was to: (1) establish the restorable yield of these bearings and (2) demonstrate the technical and economic feasibility of restoring by grinding 50 sets of the three bearing types.

The bearings which are described in detail in reference 4 were 150 each 210-size (50-mm bore) split-inner-ring ball bearings made from AISI 52100, 121 each 111-size (55-mm bore) cylindrical roller bearings made from AISI M-50, and 86 triplex 7216-size (80-mm bore) angular-contact ball bearing sets.

Of the 150 split inner-ring ball bearings inspected, 145 or 96.7 percent were considered to be canidates for restoration. The five bearings which were considered to be nonrestorable had defects such as corrosion, fatigue spalls, or critical dimensions which were sufficiently out of tolerance which would not allow them to be chrome plated and reground. However, only one bearing ring fell into the later category. By considering individual components rather than complete bearings, 98.2 percent of the components were considered capable of being restored. Fifty of the 145 bearing considered restorable were chosen at random for the restoration process.

The 111-size cylindrical roller bearings were manufactured by two separate vendors. Fifty-four were from one manufacturer and 67 from a second. Of the bearings from each manufacturer, 96.3 and 92.5 percent (52 and 62 bearings),

respectively, were determined to be restorable. Based on restorable components, 96.7 percent recovery can be obtained for this bearing group. Of the bearings rejected for restoration, seven bearings had defects such as previously described for the 210-size split-inner-ring ball bearings. The bearings chosen for restoration were all from the 62 bearings from one manufacturer.

The triplex 7216-size angular - contact ball bearing sets were also from two manufacturers. There were a total of 86 sets sent to ITI. Of these, 71 sets were from one manufacturer and 15 sets were from a second. (All the sets were not complete resulting in 255 instead of 258 bearings.) Of the 255 bearings, 207 or 80.2 percent were considered capable of being restored. However, on the basis of individual components restorable, the yield would increase to 91.6 percent. These bearings evidenced corrosion and rusting of the raceways subsequent to removal from operation. The appearance of these bearings would suggest that caution be taken at the overhaul facility to protect the bearings from corrosion after their inspection and rejection. The triplex bearings had similar damage to those reported for the other two types of bearings. Spalling fatigue failure occurred in 7 or 8.1 percent of the triplex sets. The 50 triplex sets chosen for restoration were from two manufacturers.

Based upon inspection of the three bearing types, the restorable yield was greater than 90 percent. While this number coincides with the original prediction for bearing restoration, this yield may vary with different engine and transmission applications. It is therefore recommended that careful statistics be kept to more accurately determine the bearing restoration yield for different bearings and applications over an extended time period. These statistics would aid in AVSCOM logistic and economic planning. The information would also aid the prospective bearing vendor in determining a reasonable price quotation for bearing restoration by grinding.

Unit Cost Savings of Restoration by Grinding

Restoration by grinding can decrease costs to the military services by salvaging most of those bearings which current refurbishment programs cannot restore to usable bearings. Another major advantage of restoration by grinding is that this process restores bearings to original manufacturing accuracy because restoration is done by grinding on the same machines and by the same techniques as new

bearings. This latter factor assumes greater importance when considering that today's jet engines and aircraft transmissions demand a level of bearing accuracy higher than those designed five years ago. Tomorrow's engines, now on the drawing board, reveal the need for even greater complexity bearings with a corresponding increase in bearing cost. It will be more difficult to refurbish bearings by conventional means, and the only viable alternative will be the restoration by grinding process.

Some of the tolerences of ball and roller bearings required to operate in the 2 to 3 million DN range for these engines now starting production are as follows:

Roller Bearings:

Raceway concavity/convexity = 0.000015 inch

Raceway surface finish = 4 μin. rms

Raceway deviation from true plane = 0.000020 inch

Ball Bearings:

Raceway finish = 2 μin. rms

Racewas curvature accuracy = 0.0001 inch

When these types of accuracies have to be maintained then the restoration by grinding process will prove to be not only necessary, but in terms of overall cost, the most economical means of reclaiming used bearings.

Table II lists eight bearings ranging from conventional designs of today to very complex designs for tomorrow's aircraft. It shows the unit price if these bearings are purchased new in some economic lot quantity and the estimated cost to restore by grinding in the same lot quantity. The last column in the table illustrates that the savings created by bearing restoration increases as bearing complexity increases. The relatively simple accessory roller bearing shown in figure 7 can be restoree at 47 percent of new bearing cost; today's production engine main shaft bearings (figs. 2 and 4) at 45 percent; advanced jet engine bearings now going into production at 38 percent; and for future engine designs, the figure drops to 27 percent. As bearing prices increase, the total dollar savings become, of course, even more significant than the percentage figure.

Another important factor not directly considered in the economic analysis is the potential savings in raw material. For the UH-1 helicopter it is estimated that approximately 68 800 pounds of steel can be saved per year through bearing restoration (table III). In addition approximately 5800 pounds of critical alloying elements such as molybdenum, manganese, chromium, nickel and vanadium can

be saved (table IV).

LOGISTICS OF A RESTORATION BY GRINDING PROGRAM

Restoration by grinding must be accomplished by the same processes and on the same machines as is the manufacture of new bearings. Furthermore, the same production and manufacturing planning must be employed. This necessitates that a minimum economic lot quantity of parts be processed through the plant for restoration by grinding in order that the price for restoration be maintained (table II). This economic lot quantity is approximately the same as is required for new aircraft bearing manufacture. The economics inherent in performing a given operation on a minimum quantity of parts applies whether a new raceway is ground in a newly manufactured bearing or regrinding the race in the restoration by grinding process. Similarly, each machine and inspection set-up must be capable of being amortized over a given minimum quantity of bearings.

This economic lot quantity for restoration by grinding varies with bearing size and complexity. A complex mainshaft bearing such as shown in figures 2 and 4 may have an economic lot quantity of 200 to 300 bearings while a less complex accessory bearing of the type shown in figures 7 and 8 may have 400 to 500 economic lot quantities.

It is recognized that the quantities of bearings removed at even a series of engine or transmission overhauls, will not necessarily yield an economic lot quantity of bearings. It is, therefore, necessary to store such bearings either at the overhaul base or send them piece meal to the manufacturer who is to conduct the restoration by grinding. When the predetermined economic lot quantity of a given bearing size has been accumulated, the process of restoration by grinding can be started.

The manufacturer inspects bearings as received and makes the disposition by bearing size and part number as to which bearings and/or components of bearings, i.e., inner and outer rings and separators, are recoverable by restoration. Good parts are then stored in a parts bank and bad parts scrapped. A record by bearing part number is maintained. After the required number of restorable components, including a scrap allowance of 5 to 10 percent, has been accumulated, restoration by grinding begins according to process travelers and inspection points written in much the same manner as for new bearings.

An overview of the logistics of restoration by grinding is shown on the flow chart in figure 9. It outlines the path followed from the point at which bearings are sent to the bearing manufacturer to the point where restored bearings are returned.

REQUEST FOR QUOTATION

Although table II gives a general range of savings which restoration by grinding can achieve, the exact savings must be determined on the basis of each individual bearing design. Certain characteristics peculiar to an individual bearing design may result in larger or smaller savings.

After the bearing manufacturer has reviewed the print, prices will be quoted based on lot quantity to be restored, starting with the minimum economic quantity that will assure a minimum restoration by grinding price. Lot prices for larger quantities may show further savings.

The manufacturer must also review, by physical inspection, the average condition of bearings removed at overhaul, to determine the potential yield rate of restoration by grinding. It is also necessary that an engineering evaluation be made to determine how critical is the bearing application. This can have an impact on the restoration by grinding process employed. There is a difference between newly manufactured mainshaft and accessory bearings in terms of accuracy and inspection level required.

The lot quantity on which the manufacturer is asked to quote should correspond to the rate at which bearings are removed and become available for restoration by grinding. The delivery of bearings restored by grinding will, of course, depend on this rate of availability. The total restoration by grinding process will take approximately 22 weeks after a production lot has been started.

The overall logistics necessitates that long range commitments be made. The longer the commitment and, therefore, the more total bearings are to be restored, the lower the quoted price for restoration by grinding because it will then be possible to amortize start-up and set-up costs over a greater number of units.

The quoted price will be fixed for the period of time specified. Parts will be stored in parts bank at no charge to customer. Any component or bearings scrapped, for whatever reason, will be done at no charge to either customer or manufacturer.

Initial start-up time from signing of a purchase order to the first delivery

of restored bearings is six months. Table V summarizes the quotation or contract negotiation steps of restoration by grinding.

SUMMARY OF RESULTS

A study was initiated by the Army Aviation System Command in conjunction with the NASA Lewis Research Center and the Corpus Christi Army Depot for the Industrial Tectonics, Inc. Bearing Division to undertake a pilot program to determine the economic impact and the manner in which rolling-element bearing restoration by grinding can be implemented and operated; to establish the restorable yield of the restoration process and to determine the technical feasibility of restoration by grinding on 50 sets of three bearing types. The investigation was conducted on 529 bearings comprising three different types removed at overhaul from UH-1 helicopter engines and transmissions. The following results were obtained.

(1) On the basis of individual bearing components, greater than a 90 percent yield can be achieved by the restoration process. This yield may vary with different engine and transmission applications.

(2) The estimated cost to restore rolling-element bearings by grinding ranges from 27 to 47 percent of new bearing cost depending on bearing size and complexity. The more complex a bearing is, the greater the savings achieved by restoration.

(3) The lot quantity for bearing restoration should correspond to the rate at which bearings are removed from operation and become available.

(4) Initial start-up time from signing of a purchase order to the first delivery of restored bearings is six months.

REFERENCES

1. Stanley, D. C., "Bearing Field Inspection and Refurbishing," Paper presented at Symposium on Propulsion System Structural Integration and Engine Integrity, Naval Post Graduate School, Monterrey, Calif., Sept. 36, 1974.

2. Bamberger, E. N., et al., "Life Adjustment Factors for Ball and Roller Bearings - An Engineering Design Guide," ASME, New York, 1971.

3. Bamberger, E. N., Zaretsky, E. V., and Signer, H. "Endurance and Failure Characteristics of Mainshaft Jet Engine Bearings at $3x10^6$ DN" ASME Paper no. 76-LUBS-16, 1976.

4. Parker, R. J., Zaretsky, E. V., and Chen, S. M., "Evaluation of Ball and Roller Bearings Restored by Grinding," Proc., Bearing Restoration by Grinding Seminar, AVSCOM, St. Louis, May 20-21, 1976.

TABLE I. - REJECTION RATE OF BALL AND ROLLER BEARINGS OVER 50 MM BORE REMOVED AT ENGINE AND TRANSMISSION OVERHAUL FROM UH-1 HELICOPTER[a]

	Qty brgs rejected over 3/mo period	Rejection rate	Failed from fatigue	Restorable
From turbine engine	1800	56%	180	1620
From transmission	2412	47%	240	2172
Total	4212		420	3792

[a]Source: Letter April 1972; Director of Maintenance; U.S. Army Aeronautical Depot Maintenance Center, Corpus Christi, Texas.

TABLE II. - BEARING COST COMPARISON: .

NEW VS RESTORATING BY GRINDING

Cost	New	Restored	Percent of new
Accessory roller brg (fig. 7)	$115	$65	57
Accessory roller brg (fig. 8)	$165	$75	45
Propeller thrust ball bearing-turbo-prop (fig. 10)	$210	$95	45
Mainshaft ball brg today's production jet engines (fig. 2)	$375	$170	45
Mainshaft roller brg today's production jet engine (fig. 4)	$450	$200	45
Mainshaft roller brg advanced jet engine (fig. 11)	$850	$325	38
Mainshaft ball brg advanced jet engine (fig. 12)	$1075	$410	38
Mainshaft ball brg future engine design (fig. 13)	$4500	$1200	27

TABLE III. - RAW MATERIAL SAVINGS FOR UH-1 HELICOPTER

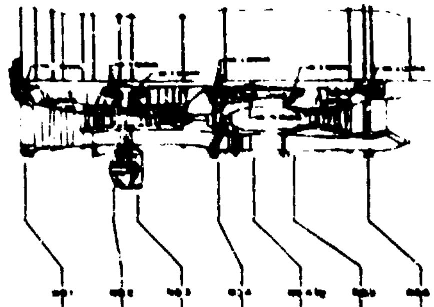
Material	Estimated bearing restored	Average bearing weight (lbs)	Recycled steel (lbs/year)
AISI M-50 steel	6 400	8	51 200
AISI 52100 steel	8 800	2	17 600

TABLE IV. - CRITICAL ALLOYING ELEMENT SAVINGS FOR UH-1 HELICOPTER

Material recycled	Total raw material from table III, lbs	Alloying Element, lbs (%)				
		Molybdenum	Manganese	Chromium	Nickel	Vanadium
AISI M-50	51 200	2432 (4.75)	-----	2304 (4.5)	------	666 (1.3)
AISI 52100	17 600	------	70 (0.40)	270 (1.5)	70 (0.40)	----
Total lbs/yr	68 800	2432	70	2574	70	666

TABLE V. - REQUEST FOR QUOTATION

Identification of bearings	Engine or transmission manufacturer's part number Federal stock number OEM part number
Technical data	Furnish drawing to which bearing is inspected Description of application Special restoration by grinding allowed or desired
Quantity	Estimate of annual quantity of bearings available for restoration and length of contractual commitment Estimate of monthly shipments to manufacturer Expected rate of delivery of bearings restored by grinding
Manufacture's response	Price quotation per bearing size Delivery time of restored bearing



	No. 1	No. 2	No. 3	No. 4	[illegible]	No. 5	No. 6	TOTAL
ANNUAL REPLACEMENT AT OVERHAUL	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
TOTAL REPLACEMENT COST	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
TOTAL REFURBISHING COST	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
ANNUAL SAVINGS								[illegible]

Figure 1. - Refurbishing experience on the JT8D engine mainshaft bearings only.

E-6726a

Figure 2. - Jet engine ball bearing.

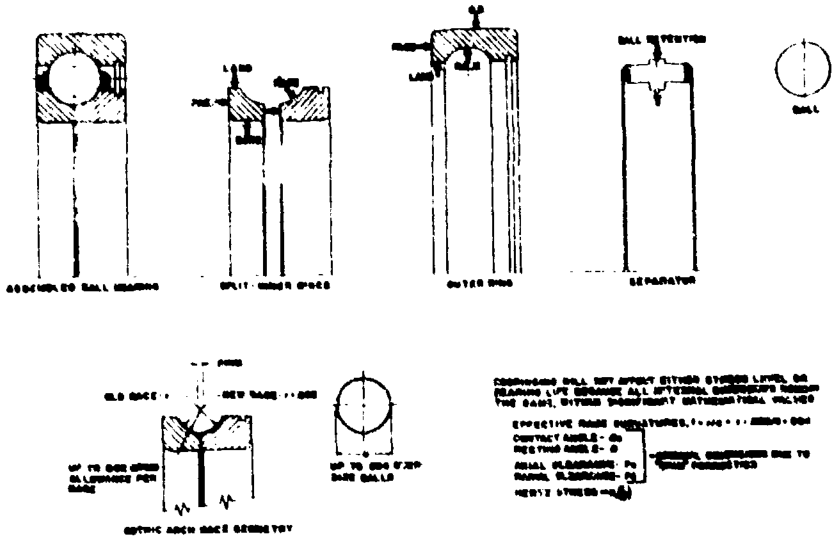


Figure 3. - Refurbishing of split inner ring ball bearing.

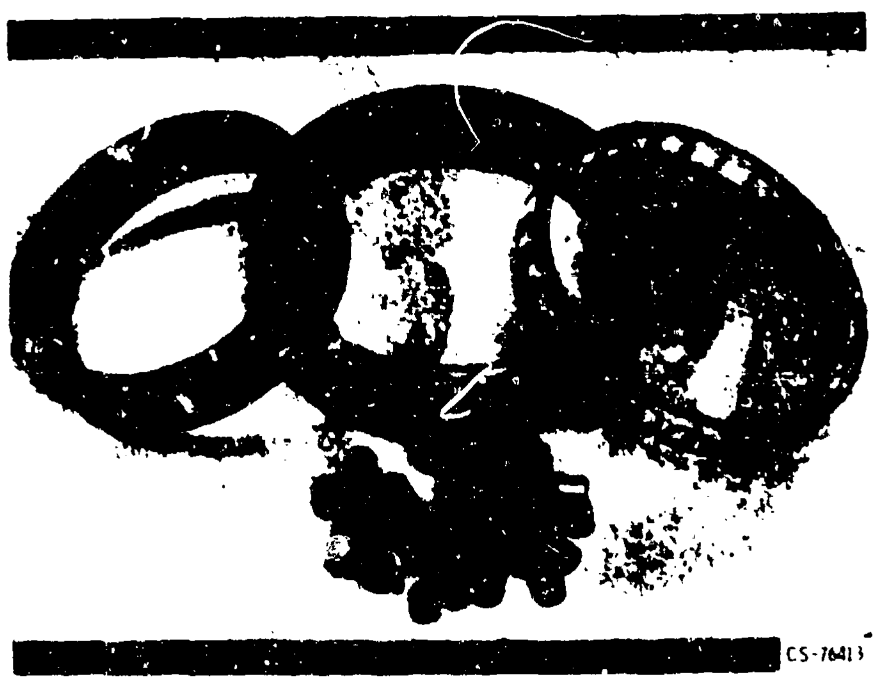


Figure 4. - Jet engine roller bearing.

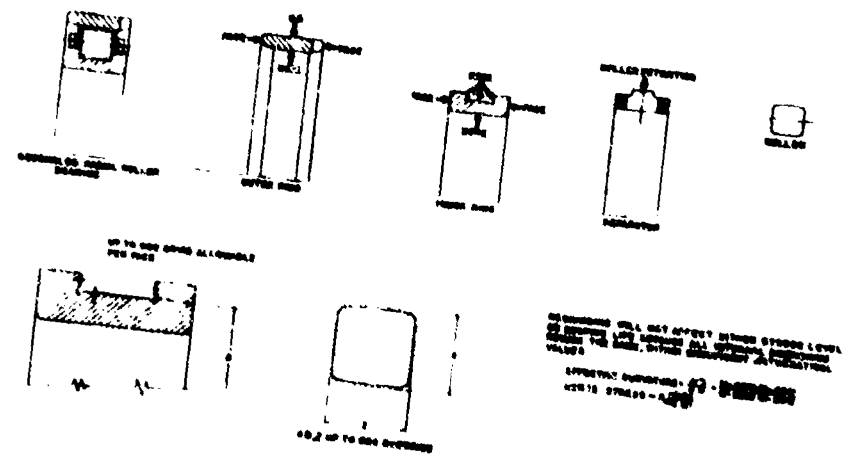


Figure 5. - Refurbishing of radial roller bearing.

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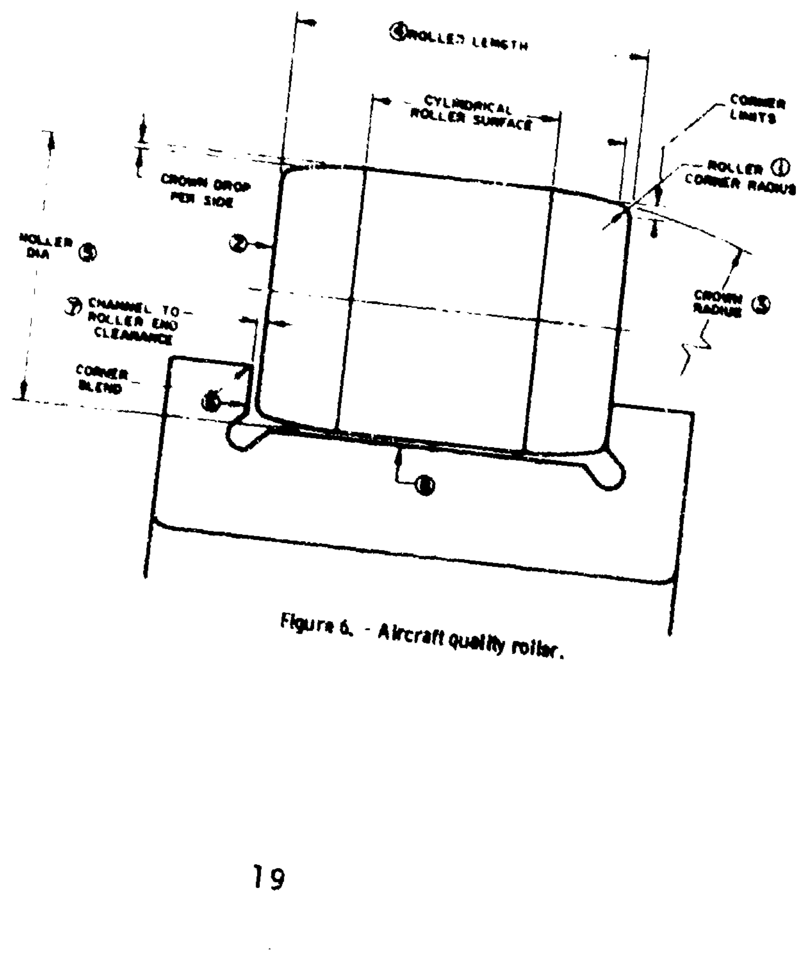
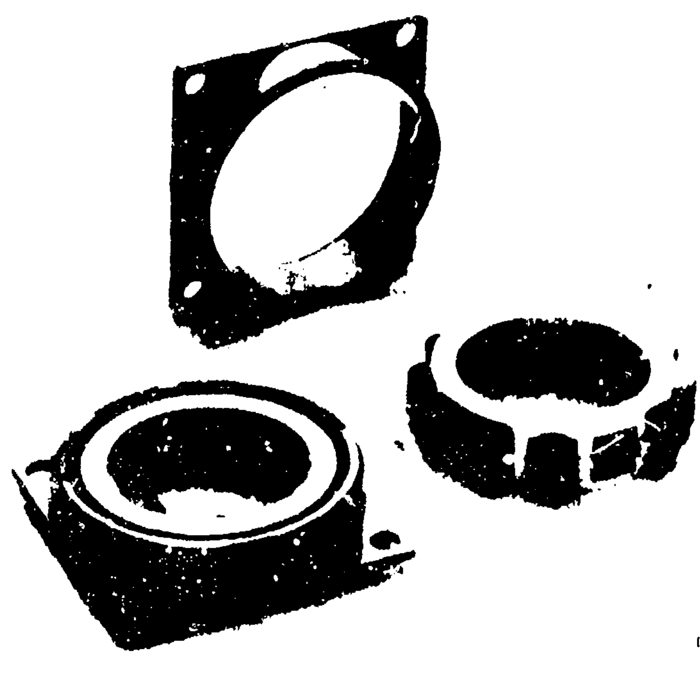


Figure 6. - Aircraft quality roller.

CS-76412

Figure 7. - Transmission roller bearing.



CS-76417

Figure 8. - Internal flange transmission roller bearing.

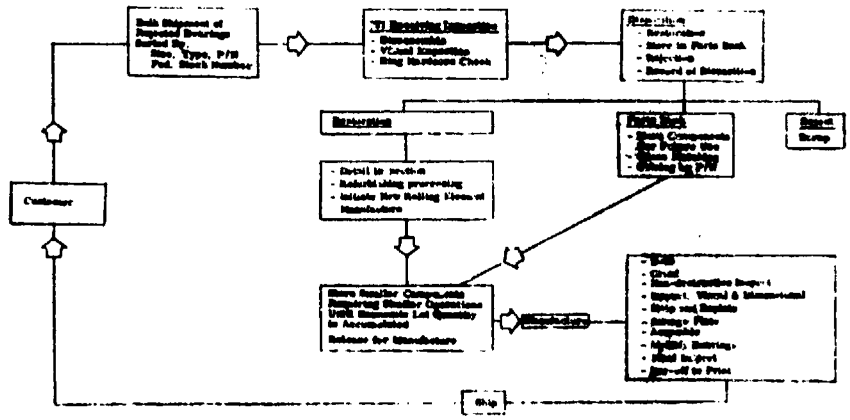
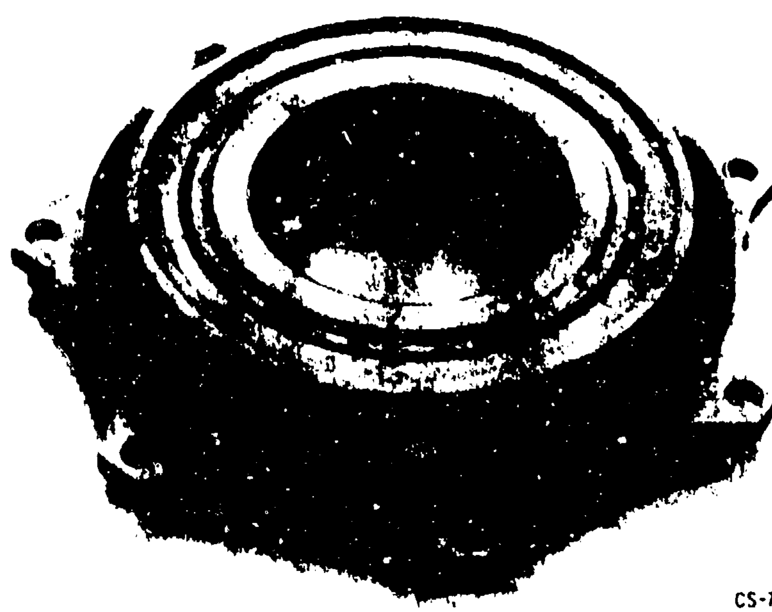


Figure 9. - Logistics of a restoration by grinding program.



CS-76418

Figure 10. - Propeller thrust ball bearing.

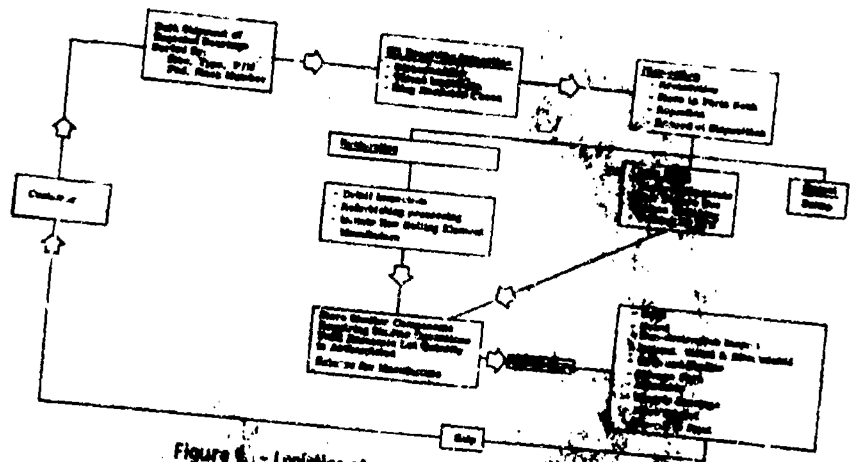


Figure 9. - Logistics of a restoration by grinding program.

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Figure 10. - Propeller thrust ball bearing. CS-76418

CS-76416

Figure 11. - Advanced jet engine roller bearing.

CS-76415

Figure 12. - Advanced jet engine mainshaft ball bearing.

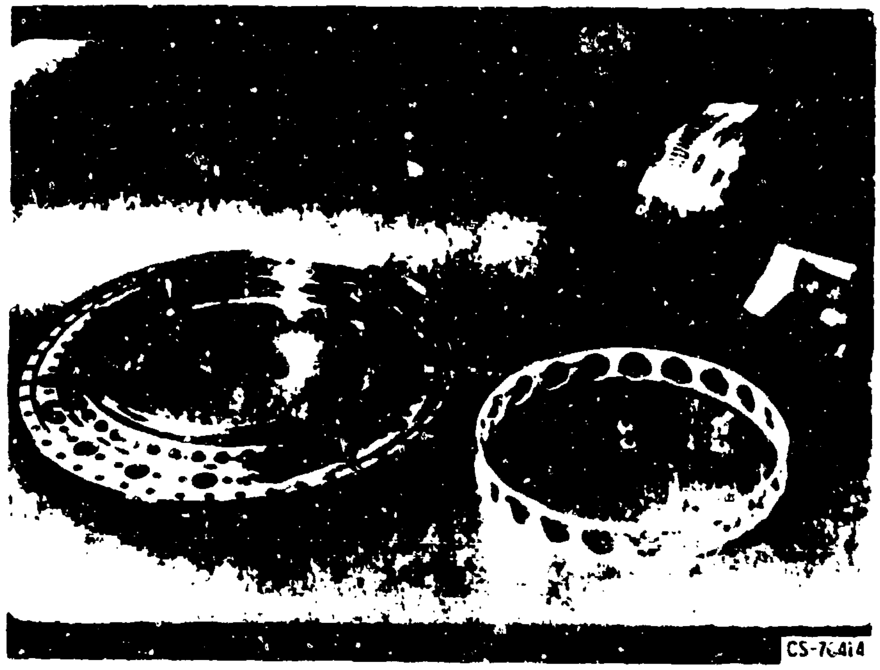

Figure 13. - Future jet engine mainshaft design ball bearing.

NASA-Lewis

E-8728b

EVALUATION OF BALL AND ROLLER BEARINGS RESTORED BY GRINDING

by R. J. Parker, E. V. Zaretsky, and S. M. Chen*

NASA Lewis Research Center

ABSTRACT

A joint program was undertaken by the NASA Lewis Research Center and the Army Aviation Systems Command to restore by grinding those rolling-element bearings which are currently being discarded at aircraft engine and transmission overhaul. Three bearing types were selected from the UH-1 helicopter engine (T-53) and transmission for the pilot program. Groups of each of these bearings were visually and dimensionally inspected for suitability for restoration. A total of 250 bearings were restored by grinding. Of this number, 30 bearings from each type were endurance tested to a TBO of 1600 hours. No bearing failures occurred related to the restoration by grinding process. The two bearing failures which occurred were due to defective rolling elements and were typical of those which may occur in new bearings. The restorable component yield to the three groups was in excess of 90 percent.

INTRODUCTION

The last three decades have seen a significant increase in the severity of applications in which rolling-element bearings are expected to function reliably and with long life. Rolling-element bearings are now required to operate at much higher speeds and, to a lesser extent, higher temperatures than in the early 1940's. The increased speed and temperature requirements originated principally with the advent of the aircraft gas turbine engine. Its development, coupled with the appearance of a variety of high-speed turbine-driven machines, has resulted in a wide range of rolling-element bearing requirements for main-shaft, accessory and transmission applications.

Classical rolling-element fatigue which is of subsurface origin has been considered the prime life limiting factor for rolling-element bearings although

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actually less than 10 percent of them fail by fatigue. With proper design, handling, installation, lubrication, and system cleanliness, a rolling-element bearing will eventually fail by fatigue. Because fatigue results from material weaknesses, research to improve material quality has been a continuing activity. The remaining 90 percent of the failures are due to causes such as lubricant flow interruption, lubricant contamination, lubricant deterioration, excessive dirt ingestion, improper bearing installation, incorrect mounting fits, mishandling of bearings prior to installation, installing a contaminated bearing, manufacturing defects, ring growth in service, and corrosion.

Nonmetallic inclusions are one cause of classical rolling-element fatigue (refs. 1 to 5). Basic inclusion types include sulfides, aluminates, silicates, and globular oxides. These inclusions may act as stress raisers similar to notches in tension and compression specimens or in rotating beam specimens. Incipient cracks emanate from these inclusions (fig. 1), enlarge and propagate under repeated stresses forming a network of cracks which form into a fatigue spall or pit (fig. 2). In general, the cracks propagate in a plane approximately 45° to the normal; that is, they appear to be in the plane of maximum shearing stress. In addition to nonmetallic inclusions, large carbides can act as stress raisers and nucleate fatigue cracks (refs. 6 and 7).

One method for increasing rolling-element bearing life, reliability and load capacity is to eliminate or reduce nonmetallic inclusions, entrapped gases, and trace elements. Melting steel in a vacuum provides large life improvements (refs. 8 to 10). Double vacuum-melted bearing steel, now commercially available, is processed with the first heat being vacuum induction melted (VIM). The material is subsequently vacuum arc remelted (VAR). The result is a material with marked reductions in nonmetallic inclusions, gas content, and trace impurities (ref. 11). Tests with 120-mm bore ball bearings made from VIM-VAR AISI M-50 steel produced fatigue lives at least seven times that achieved by vacuum arc remelted steel (ref. 12). Hence, the probability of subsurface fatigue can be greatly minimized within current TBO intervals and bearing steel state-of-the-art.

Failure by the other modes enumerated above are for the most part nonpredictable and tend to be surface as opposed to subsurface originated. In general, these failures due to surface originated defects, occur much earlier than those failures due to classical rolling-element fatigue (refs. 13 to 15). As a result, in aircraft engine and transmission applications, a large number of bearings are discarded at overhaul or during periodic maintenance. This

results in millions of dollars in bearings which have to be replaced anually. In addition, critical alloying elements such as molybdenum, manganese, chromium, nickel, and vanadium are consumed.

In order to reduce costs and conserve materials by reusing bearings, bearing rework has been performed both in private industry and within the U.S. Government on a limited basis. Basically, bearing rework as currently practiced comprises the following steps (ref. 16).

1. Replace defective balls or roller sets with new balls or rollers that match the original parts with respect to size, tolerance, finish, sphericity, hardness, and material. The bearings are then reassembled and checked for radial and axial clearance and, where required, contact angle.

2. Defective retainers are replaced. Alternately, silver plating from original retainers are removed, and the retainers are nondestructively inspected and replated to original specifications.

3. Ball and roller raceways are polished to remove superficial corrosion or other shallow surface defects. The reworked bearing surfaces are then checked for profile, concentricity, surface finish, and metallurgical integrity.

4. The inner- and outer-diameter surfaces of the bearing rings are ground and replated to original drawing specification.

5. Duplex and multiple stack bearing sets are rematched in accordance with original specifications using individual bearings from rejected or failed sets.

The aforesaid method retains subsurface damage in the used bearing raceways which may propagate to fatigue failures and thus limit the reworked bearings' life and reliability. Reworking of any kind which involves significant removal of material (and thus this damage) from bearing raceways has been avoided in previous refurbishing or reworking techniques.

The program reported herein was undertaken to investigate the endurance characteristics of aircraft turbine engine and transmission bearings whose raceways were restored by grinding. The primary objectives were to (1) determine the yield of bearings suitable for restoring by grinding; (2) determine the endurance characteristics of restored bearings under simulated operating conditions; (3) determine post-test condition of the restored bearings; and (4) establish a specification for bearing restoration by raceway grinding.

BEARING SELECTION FOR RESTORATION

Three bearing types were selected from the UH-1 helicopter engine and transmission for the program of restoration by grinding. The criteria for

selection were (a) bore of 50-mm or larger and (b) acquisition cost to the Government of $50.00 or greater. Based on information furnished by the Corpus Christi Army Depot, the bearings selected had among the highest replacement rates during normal maintenance and overhaul. The bearings and their Federal Stock and component engine part numbers are listed in table I.

The 210-size (50-mm bore) split inner-ring ball bearing is shown in figure 3. These bearings are made from AISI 52100 steel by a single manufacturer. The bearing cage or separator which is made from silver-plated bronze is of a one-piece design and is inner-land riding. These are mainshaft bearings from the front compressor position of the T-53 gas turbine engine. Engine operating conditions for the bearing include a speed of 24 000 rpm and a thrust load of 450 pounds. Bearing oil-in temperature in the engine is approximately 200° F and oil-out temperature is between 250° to 300° F. A total of 150 of these bearings were sent by the overhaul depot to be inspected for restoration.

The 111-size (55-mm bore) cylindrical roller bearings shown in figure 4 are manufactured by two separate manufacturers. The design of each manufacturer is sufficiently different whereby interchangability of the bearing components is not possible. The bearings are manufactured from AISI M-50 steel. The bearing cage or separator is a one-piece, silver-plated steel, inner-land riding design. These bearings are from the T-53 gas turbine engine rear compressor position. Operating loads are nominal for this bearing. They are considered to be under a light radial load. The maximum bearing speed is 24 000 rpm. Bearing oil-in temperature in the engine is approximately 200° F and oil-out temperature is approximately 350° F. A total of 121 of these bearings were sent by the overhaul depot to be inspected for restoration.

The third bearing type was the triplex ball bearing set shown in figure 5. Eighty-six sets of these bearings were sent to be inspected for restoration. These are 7216-size (80-mm bore) angular-contact ball bearings which are mounted on the input bevel gear pinion shaft of the UH-1 helicopter transmission. The bearings are manufactured from AISI M-50. They have a one-piece, silver-plated steel, inner-race riding cage. Operating conditions in the helicopter for the triplex set include a radial load of 3175 pounds and a thrust load of 4274 pounds at a speed of 6600 rpm. All the bearings in the set are match ground with a 100-pound preload. There are two manufacturers of this bearing. However, each manufacturer's design is similar which allows for complete interchangeability.

INSPECTION OF REMOVED BEARINGS

A total of 529 bearings of the three types were subjected to inspection to determine suitability and yield for restoring by grinding. These included 150 split inner-ring ball bearings (210-size), 121 cylindrical roller bearings (111-size), and 86 triplex sets of the angular-contact ball bearings (7216-size). The inspection procedure consisted of visual and dimensional inspections including ring diameters, flatness, roundness, hardness, magnaflux for flaws, raceway surface conditions (spalling, wear, etc.), and general suitability of all components for restoring by grinding. The restorable bearing yield percents are given in table II for the three bearing types.

Of the 150 split-inner-ring ball bearings inspected, 145 were judged restorable for a yield of 96.7 percent. On the basis of restorable components, the yield was 98.2 percent. The general appearance of this group of bearings was excellent. Raceway tracks were light and balls and separators were in good condition except for some oil varnish deposits on separator lands and ball pockets. The five bearings that were judged non-restorable had defects including fatigued raceways, corrosion, or bore and/or outside diameters (O. D.) greater than 0.001 inch out of tolerance. The incidence of bore and O. D. variance from print tolerance was high at 47.3 percent. Some of this variance is explainable by gaging discrepancies in cases were 0.0001 inch off-print dimension was noted. However, 28 of these bearings had rings out of tolerance from 0.0002 to 0.001 inch but were judged restorable since these bore and O. D. surfaces could be plated and reground to print dimensions. Only one bearing was non-restorable solely because of out-of-tolerance rings. Fifty of the 145 restorable bearings were randomly chosen to be restored by grinding.

These bearings are made of AISI 52100 steel, which may be expected to experience dimensional changes in turbine-engine mainshaft positions where soak back temperatures near 400° F may be expected. Generally, the use of AISI M-50 steel for mainshaft bearings is recommended.

The 121 cylindrical roller bearings consisted of 54 from manufacturer A and 67 from manufacturer B. The number of bearings judged restorable were 52 and 62, respectively, for yields of 96.3 and 92.5 percent, respectively, and an overall yield of 94.2 percent. On the basis of restorable components, the yield was 96.7 percent. The general appearance of these bearings was excellent. Raceway tracks were light, and the rollers and separators were in good condition except for some oil varnish deposits on the separator lands and

roller pockets. The seven bearings that were judged non-restorable had defects including fatigue spalling, corrosion, or surface distress on the raceways and damaged O. D. surfaces of outer rings. Seventeen of the total group of 121 bearings had bore or O. D. dimensions out of print tolerance, but only three of these discrepancies were greater than 0.0001 inch.

The bearings chosen for restoring by grinding were 50 randomly chosen bearings from the 62 restorable bearings all from manufacturer B.

The total of 86 triplex sets of the angular contact ball bearings consisted of 71 sets from manufacturer C and 15 sets from manufacturer D. The number of sets judged restorable were 46 and nine, respectively, for yields of 64.8 and 60.0 percent, respectively, and an overall yield of 64.0 percent. The total number of bearings considered restorable was 207, for an overall bearing yield of 80.2 percent. On the basis of components, that is, rings and cages, the overall restorable component yield was 91.6 percent.

The general appearance of this group of bearings was much poorer than the other two groups of bearings inspected. There was a high incidence of corrosion and rusting in the raceways. Most of this apparently occurred after removal from the transmission and could be attributed to insufficient protection during or prior to storage. Many of the raceway tracks appeared to have experienced wear or a change in surface characteristics associated with operation in dirt or debris contaminated lubricant. The incidence of spalling fatigue failures was much higher than with the other two groups. Some of this spalling may have been related to debris damage.

Of the 48 bearings that were judged non-restorable, 37 had corrosion pitted or rusted raceways, two had separator damage, and nine had raceway spalling fatigue failures. Three bearings from two triplex sets were not returned for inspection by the overhaul depot, thus accounting for the discrepancy in quantities from the total of 86 triplex sets. The spalling fatigue failures occurred in seven triplex sets, for an 8.1 percent failure rate. Bearings from 24 triplex sets or 27.9 percent had corrosion pitted or rusted raceways.

Only 25 of the 255 bearings inspected had bore or O. D. dimension discrepancies greater than 0.0001 inch, but none of these were bad enough to be judged non-restorable.

The 50 triplex sets chosen for restoring by grinding included sets from both manufacturers C and D.

The restorable yield on the basis of components (rings and separators) was greater than 90 percent for each bearing type inspected. For the split-inner-ring

ball bearings the restorable component yield was 98.2 percent; for the cylindrical roller bearing, 96.7 percent; and for the triplex angular-contact ball bearings, 91.6 percent.

PROCESS OF RESTORING BY GRINDING

Figure 6 shows the relative magnitude of shearing stress as a function of depth below the surface. In the zone of maximum resolved shearing stress, which generally occurs from 0.002 to 0.006 inch below the surface for most rolling-element bearing applications, there exists the maximum orthogonal shearing stress, the maximum octahedral shearing stress, and the maximum shearing stress on a 45° plane. Due to the cyclic nature of the shearing stresses, a fatigue crack will eventually occur within this zone and propagate into a crack network. This network will eventually lead to spalling and failure of the raceway bearing surface whereby the bearing is no longer suitable for the purposes for which it was intended.

In addition to this failure phenomena, damage to the surface can occur which is caused primarily by dirt and foreign object or wear debris within the lubrication system and in some instances by corrosion of the raceway surfaces. This surface damage, in the form of indentations or scratches, acts as stress raisers on the bearing raceway surfaces, and also initiates fatigue cracks which propagate into crack networks and eventually leads to spalling.

Where a rolling element such as a ball or a roller fails by spalling, debris damage generally occurs to the raceways. In addition, where an inner or outer raceway fails there is generally debris damage to the rolling elements and the non-failed race. These failure phenomena make the bearing unusable both for continuous application and for subsequent reuse. However, most of the debris damage occurs at depths less than 0.002 inch below the surface of the rolling-element raceways.

Another problem in the operation of rolling-element bearings is growth of the bearing race rings. As an example, the inner or outer races, can grow due to metallurgical transformations or due to hoop stresses during operation. This growth results in the bearing being not reusable after removal from its application.

For aircraft applications, bearings removed at overhaul are usually cleaned and visually inspected for defects and dimensional conformance to print.

Bearings are usually not taken apart, except those which by design are separable. Therefore, impending or minute fatigue spalling cannot be detected except by hand, feel, or noise testing. Both of the latter techniques are very subjective and can only be used with any degree of success by a bearing manufacturer and only after establishing performance yardsticks on large production runs of new bearings.

In general, the rejection rate of bearings removed from turbine engines and transmission at overhaul is approximately 50 percent. Based upon experience, less than 10 percent of the bearings removed have failed due to fatigue. Therefore, 90 percent of those rejected bearings can be restored by grinding the raceways and thus reused since the raceway damage causing rejection is usually much more superficial than fatigue failure damage.

In order to restore bearings economically, it is necessary to start a production run of components when a minimum economic lot size of components has been accumulated. A lot is then processed utilizing process-travelers and manufacturing drawings in same manner as a manufacture of new bearings. Some quantity of bearings received for restoring have non-restorable components. These are scrapped. The restorable components are stored in a "parts bank" from which interchangeable components will be drawn to make an economic lot size for restoring.

Bearings rejected for reuse in application are disassembled into its component parts. These components are visually inspected, and the hardness of the bearing races are measured. The bearing components are either put aside for restoring or scrapped.

Those components determined to be restorable are dimensionally inspected. Where necessary, the bearing faces, bores, and outer diameters are ground and either nickel or chrome plated to a thickness that will allow the surfaces to be reground to the original print dimensions.

Both inner and outer raceways are ground to a depth not exceeding the maximum depth of the maximum resolved shearing stress under their maximum loaded condition but not less than 0.002 inch. The surfaces finish is maintained to its original print specification. The bearing is then refitted with new rolling elements of a diameter equal to the diameter of the elements previously contained in the bearing plus twice the depth of regrinding.

For ball bearings the effective race curvature is identical to the original dimensions within significant mathematical values. The original values of contact angles, resting angle, and radial clearance remain unchanged. Although

the restored bearing contains oversize balls and oversize raceways, the total effective geometry of the bearing has not been changed, and consequently, the contact stress level and calculated bearing life of the restored bearing will be essentially identical to that of the original bearing. The bearing separator is stripped of its silver plating, where applicable, inspected for cracks and replated. The new, oversized rolling elements are placed within the separator, and the bearing is reassembled.

For cylindrical roller bearings the procedure as outlined above is the same with the exception that the roller length as well as the roller diameter are increased by a value twice of the depth of regrinding.

Referring to figure 6, the location of the zone of maximum resolved shearing stress in the inner and outer raceways after regrinding is displaced by a distance X, and the stress is redistributed accordingly. The dimension X should be not less than 0.002 inch nor more than the original maximum depth of the zone of maximum resolved shearing stress at the maximum load condition for the bearing. The bearing kinematics, internal clearances, and contact loads during operation remain unchanged. A new volume of material is being stressed which should result in a life or probability of survival equivalent to the bearing's original life or survival probability.

ENDURANCE TESTING

Endurance tests were performed in order to evaluate the process of restoring by grinding on each of the three bearing types and determine that the restored bearing will provide lives at least as long as the desired time between overhaul of 1600 hours. Speed, load, and lubrication conditions were chosen to be representative of each bearing application. The test conditions are shown in table III.

Test Apparatus and Procedure

Each of the bearing types were tested in test heads specifically chosen for the particular speed and load conditions. The test facilities were capable of continuous running with test interruption only due to bearing failure or inadvertent test facility malfunctions. The lubrication systems for the three facilities had many common features. Each system used MIL-L-23699 Type II ester, from single lubricant batches. The test bearings were lubricated by jets.

The lubricant was recirculated through appropriate heat exchangers to maintain the desired flow rates and lubricant-in temperatures.

In order to simulate lubricant replenishment due to leakage and evaporation in engine and gearbox lubrication systems and periodic lubrication changes, the test facility lubrication systems were periodically drained and refilled with new lubricant. Also, the lubricant was changed each time at new bearing or bearings were put on test.

Split-inner-ring ball bearing test head. - A sectional view of a test head used for the 210-size split-inner-ring ball bearings is shown in figure 7. Four of these test heads were used, each one driven through a quill coupling from a support spindle driven by flat belts and 25 horsepower electric motors. One of each pair of test bearings was mounted in a floating housing and was thrust loaded against the other with a calibrated hydraulic cylinder.

Lubrication is provided by jets located between the test bearings. Two test lubrication systems, separate from the support bearing lubrication systems, were used, each supplying lubricant to two test heads (four test bearings). Each test lubrication system contained a heated 4-gallon sump, a 1.7-gallon per minute at 100 psi supply pump, a nominal 10-micron absolute filter, and a turbine type flowmeter. Scavenge from the test housings was by gravity drain lines. Bearing outer-race temperature and lubricant inlet and outlet temperatures were measured by thermocouples and continuously recorded on a strip chart recorder. Oil flow rates, spindle speeds, load system pressure, and lubrication system pressures were periodically recorded. Continuous, unattended 24 hours per day, 7 days per week operation was accomplished.

Cylindrical roller bearing test head. - A section view of a test head used to test the 111-size cylindrical roller bearings is shown in figure 8. Three of these test heads were used, each containing four test bearings and two support ball bearings. The test spindles were driven by 7.5 or 10 horsepower electric motors and a geared speed increaser. Power was transmitted from motor to gear box by pulleys and V-belts. Flat belts transmitted power from the gear box to the test spindle. Each pair of test bearings was radially loaded by a spring scale and lever arm arrangement.

Lubricant was supplied to the test bearings through jets located between each pair of test bearings. Each test head has a separate test lubrication system with a heated 4-gallon capacity sump, a supply pump, a water-cooled heat exchanger, a nominal 10-micron filter, a volumetric flow meter, and a scavange pump. A separate lubrication system was used for the support bearings.

Test bearing outer-ring temperatures, oil inlet and outlet temperatures were measured with thermocouples and were continuously recorded on strip charts. Machine vibration was continuously monitored and in conjunction with several automatic pressure and temperature monitoring and safety devices, the test rigs were capable of 24 hours per day, 7 days per week, unattended operation.

Angular contact ball bearing test head. - A section view of a test head used to test the 7216-size angular-contact ball bearings is shown in figure 9. Four of these test heads were used, one on each end of two spindles driven through V-belts with 20 horsepower electric motors. A pair of test bearings were mounted in each test head in a back-to-back arrangement. The bearings were thrust loaded against each other by four bolts with a fifth strain-gaged bolt used to set and monitor the thrust load magnitude. The desired radial load was applied to each pair of bearings with a dead weight load through a 10 to 1 lever arm.

Lubrication is provided by jets located between each pair of test bearings. A single lubrication system supplied lubricant to all eight bearings tested simultaneously. This lubrication system was separate from the system supplying lubricant to the load bearings. The system for the test bearings contained a heated 12-gallon sump, a 25-gallon per minute supply pump, a constant pressure bypass system, a water-cooled heat exchanger, a nominal 10-micron full flow filter, and a 30-gallon per minute scavange pump. Lubricant temperatures were monitored by thermocouples, and flow rate was monitored by calibrated visual gages.

Continuous, unattended 24-hour per day, 7 days per week operation was accomplished utilizing a computer control system. Test bearing outer-ring temperature, thrust load, and machine vibration were continuously monitored. Lubricant inlet and outlet temperatures, oil flow rate, and spindle speed were periodically monitored.

Test Results

Thirty of each of the three bearing types were chosen at random from the groups of bearings restored by grinding. These bearings were then tested for endurance in their respective facilities for a duration of 1600 hours at conditions representative of the specific application. The objective of the tests was to demonstrate the capability of the bearings restored by grinding to operate satisfactorily for the desired time between overhaul of 1600 hours. This is in

contrast to bearing fatigue testing which is designed to run at conditions chosen to accelerate spalling fatigue failures.

In each of the three sets of tests, none of the restored bearings experienced failures which could be related to the restoring process. Twenty-eight of the thirty 7216-size angular-contact ball bearings, 29 of the 111-size cylindrical roller bearings, and all 30 of the 210-size split-inner ring ball bearing reached the desired 1600 hour time without failure.

Results with 210-size split-inner-race ball bearing. - The bearings in this group were from a helicopter turbine engine, front compressor bearing position. Eight bearings were tested simultaneously with two bearings in each of four test heads. In order to complete the 1600 hour duration for 30 bearings, the final test setup utilized only three of the four test heads. The lubricant volume in the sump was adjusted accordingly.

Subsequent to the successful completion of the 1600 hour tests, the bearings were disassembled and visually inspected. The raceways had visible running tracks typical of bearings running under thrust load. All race surfaces and ball surfaces were generally discolored from heat and lubricant staining. The raceway running tracks were discolored to a lesser extent. The condition of all contacting surfaces including raceways, balls, and cage surfaces was excellent, with no indications of any detrimental effects of the restoring process or damage from the endurance testing.

The extent of discoloration on the bearing components suggests that these bearings were exposed to relatively high temperatures in these tests. The measured outer race temperatures were in the range of 252° to 275° F and averaged about 260° F. Oil-out temperature ranged from 235° to 265° F The hardness of the inner and outer races of several bearings were measured after disassembly. These bearings included the bearing with the highest outer-race temperature (275° F) (also most discolored bearing) and the bearing with the lowest outer-race temperature (252° F). All hardnesses were in the range from 58 R_c to 60 R_c, which is on low end of the acceptable range for rolling-element bearings. These temperatures are approaching the limits for the AISI 52100 material. (It is recommended and is common practice to use AISI M-50 for all turbine engine main shaft bearings.)

Results with 111-size cylindrical roller bearings. - The bearings in this group were from a helicopter turbine engine, in the rear compressor bearing position. Twelve bearings were tested simultaneously with four bearings in each of three test heads. To complete the 1600 hour test time for all 30 test bearings, dummy bearings were used to make up the extra positions in the test head.

The measured outer-race temperatures for these bearings were in the range from 220° to 245° F. The oil-out temperature ranged from 205° to 220° F.

Twenty-nine test bearings completed the desired 1600 hour duration. One bearing suffered a failure after only 16.3 hours. Subsequent detailed examination of this bearing indicated that the failure initiated as a roller fatigue spall. Failure detection equipment did not detect the failure and shut down the test rig. It was apparent that the roller spall propagated until more than half the roller surface was severely spalled, eventually causing cage breakup and subsequent severe damage to the other rollers and the raceways. Because of the severely overrun condition of the spalled area on the suspected roller, definite evidence of a material defect was not found. However, scanning electron microscope examination of the spalled area and metallographic sections through the spalled area indicated that subsurface initiated rolling-element fatigue was the primary failure. Since the microstructure and hardness of the roller, in general, were typical of properly heat treated AISI M-50, it is suspected that a stress concentration such as an inclusion or void was in the critical area. Additionally, there was no evidence of roller skew or unusual end wear on any rollers from this bearing.

It is concluded that this premature failure was not related to the restoring by grinding process. The process includes installing new rollers in the restored bearing, and such new rollers are of a quality which would be installed in new bearings. Thus, such a failure could have occurred in a new bearing as well as in this restored bearing.

The bearings that completed the 1600 hour tests were disassembled and visually inspected. Initial examination with the unaided eye revealed the raceways in good condition, with roller tracks somewhat more apparent on the outer raceways than on the inner raceways. Cages were in excellent condition with the normal light wear or burnishing of the silver plating in the pockets and the inner race riding lands. The rollers nearly all showed some circumferential lines typically observed on tested roller bearings. The roller ends and the inner race flange contacts were in excellent condition revealing no significant abnormal roller motion. However, in a few of the bearings, the cage pocket wear indicated a very slight amount of roller skewing. However, the extent of skewing does not appear to present a problem.

More detailed examination at low magnification (6×) of the surfaces of rollers from several of the bearings revealed shallow surface distress or pitting within the flat length of the roller and generally toward the blend of the flat length

and the crown. The depth of the pitting, as measured from surface profile traces and metallographic sections, was typically less than 0.0005 inch. Although the inner and outer raceways showed some isolated evidence of very minor surface distress, the damage was mainly limited to the roller surfaces.

Surface finishes of the rollers and raceways were measured on 10 bearings randomly chosen from the total lot of 30. The outer raceways measured either 4 or 5 μin. RMS in all cases. The roller cylindrical surfaces measured from 4 to 6 μin. RMS. The inner raceway surfaces ranged from 3.5 μin. to 17 μin. RMS. The engine manufacturer's drawing for this bearing specifies 10 μin. or better for these surfaces although bearing manufacturers typically finish to better surfaces as indicated by the outer raceway and roller surfaces measured here.

In all cases where the surface finish of the inner raceway equalled or exceeded the specified 10 μin., surface distress was observed on the rollers. On two bearings, where inner raceway finishes were 6 and 7 μin. RMS, some roller surface distress was observed. On the other bearings, where inner raceways varied from 3.5 to 8 μin. RMS, no roller surface distress was observed.

The elastohydrodynamic (EHD) film thickness in the roller raceway contacts was calculated using a high-speed roller bearing computer program for the conditions of these tests. The film thickness at both the inner and outer raceway contacts is estimated to be 27 μin. An accepted criterion for the effectiveness of the EHD film thickness in given conditions with given bearing surfaces is the ratio of EHD film thickness to the composite surface roughness. This ratio is often referred to as film thickness parameter or Λ. The composite surface roughness is the square root of the sum of the squares of the surface finishes of the two surfaces in contact.

For those bearings where the inner-race surface finish was 10 μin. RMS or greater, Λ was 2.4 or less. In the worst cases, Λ was as low as 1.5. Where Λ is less than 3, it should be expected that some asperity contacts will exist (refs. 17 and 18). The detrimental effects of this surface-to-surface contacts is expected to be further aggravated by skidding. At the very low radial load of these tests, it is expected that some skidding exists, wherein the rollers are orbiting at a speed less than epicyclic speed. Under these conditions of skidding and low Λ, it may be expected that some surface damage would occur. Thus, the surface distress observed on the rollers was apparently related to the test conditions, and not attributed to the restoring by grinding process.

The load and temperature conditions for these tests are estimates of those that the bearing experiences in the engine. The engine conditions, of course, are neither constant nor easily determined. Whether the engine conditions are such that surface distress would occur, such as that observed on these test bearings, is not known, but in view of these test results, that possibility exists.

With the exception of some inner raceway surface finishes not meeting specifications, the endurance tests revealed no problems related to the restoring by grinding process. Raceway surface finish deviations are occasionally found in new bearings, so it is not a problem unique to restored bearings.

Results with 2716-size angular-contact ball bearing. - The bearings in this group were from triplex sets of a helicopter transmission input pinion bearing. The tandem pair and preload bearing arrangement is shown in figure 5. Thirty bearings were chosen at random from the tandem pairs of 30 triplex sets. Eight bearings were tested simultaneously with four bearings required for each test setup In order to complete the 1600 hour duration for 30 bearings, additional bearings were chosen at random from the remaining bearings.

The measured outer-race temperatures for these bearings were in the range from 185° to 195° F. Oil-out temperature ranged from 170° to 180° F.

After 1122 hours with one set of bearings, the inner rings of two adjacent bearings began to turn on their shaft. They were removed and could not be tested further. Their raceways, balls, and cage surfaces were in excellent condition. The bore diameters were within tolerances, but near maximum. This fact, coupled with a shaft size near minimum apparently allowed an undesirable fit situation with this particular bearing pair. Since the bores were within tolerances, this failure could not be directly attributed to the restoring process.

While running the last two bearings in the 30 bearing samples, an additional bearing, one of two which were used only as slave bearings at the opposite end of the test spindle, suffered a severe ball failure. Although this bearing was not one of the original random sample of 30 bearings, it was a bearing from the 30 restored triplex sets. Observation of the failed bearing indicated that the ball failure was due to a metallurgical defect in the ball and had no relation to the restoring process of the races.

Subsequent to the 1600 hour tests, the bearings was disassembled and visually inspected. The inner raceway had visible running tracks typical of bearings run under such conditions for extended times. The outer raceways had visible tracks typical of ball bearings under combined radial and thrust load conditions.

The general condition of all raceways, ball surfaces, and cage surfaces was excellent, with no indications of any detrimental effects of the restoring process.

General Comments

The results of the program indicate that bearing restoration is technically feasible. The experience gained from testing the three groups of bearings indicate that the performance of the restored bearings are comparable to brand new bearings. It was originally speculated that infant mortality of the restored bearings would be eliminated or at least minimized, because those bearing raceways which would have been inherently defective have been eliminated during field operation. However, the one ball failure and one roller failure experienced on this program indicate that infant mortality can still be a problem because of potentially defective new rolling elements. This problem would be no greater than that currently being experienced. In other words, while restored bearings can be expected to be comparable to newly manufactured bearings, it would be unrealistic to assume that their group life potential would be greater.

A necessary element in the bearing restoration process is to assure proper quality control of the restored bearings. Further, it is necessary to assure manufacturing consistancy comparable to newly manufactured bearings. An additional element on the bearing restoration process is the assurance that a potential vendor's restoration process will not adversely affect the life or quality of a bearing. As a result of the aforementioned AVSCOM and NASA jointly developed a proposed "Specification for Restoring Bearings by Grinding" which is included as an appendix to this report.

The specification incorporates both the technical and quality assurance aspects which have been defined both in the bearing restoration program and recent advances in rolling-element bearing state-of-the-art. The raw material and melting specifications for rolling elements calls for VIM-VAR AISI M-50 and VIM-VAR AISI 52100 assuring a longer rolling-element life potential then is currently being obtained. Minimum grinding depths are established but the exact amounts are left to the discretion of the vendor. Hardness and surface finish specifications are also tightened from previous AVSCOM practice. Research performed by the NASA indicates both factors to be critical to rolling-element bearing life and reliability. Results of the tests reported herein indicate that surface distress due to marginal elastohydrodynamic film thickness with the existing bearing surface finishes is an inherent defect in current bearing

design specification as practiced by AVSCOM bearing suppliers.

The use of AISI M-50 steel is specified in aircraft bearings instead AISI 52100 because of the improved ability of the AISI M-50 steel to retain its hardness at elevated temperatures. Unfortunately, when current bearing specifications are such as to set a minimum hardness of Rockwell C-59 at room ambient temperature, the hardness of the material at bearing operating temperatures of 250° F falls below acceptable values. This minimum hardness defeats the use of the AISI M-50 steel and can result in premature bearing damage and failure. As a result, the hardness criteria for the proposed specification is tightened from that of current AVSCOM practice.

Material and part traceability is another important factor. The ability to trace defective critical bearings in a system can be extremely important when failures are experienced in the field. Traceability is not now being practiced by the U.S. Army. A defective group of bearings or improperly processed or heat treated steel can occur both in new, refurbished, or restored bearings. Where the failure mode can cause a loss of an aircraft, it becomes extremely important to be able to remove all such defective bearings from the military aviation system. Presently, this cannot be accomplished.

Original equipment bearing vendors have a proven capability manufacturing acceptable bearings for a given application. Not all potential vendors for the restoration of aircraft quality bearings have this proven record of bearing manufacturing performance for aircraft quality bearings. This does not mean that these vendors should be excluded from the aircraft bearing restoration. As a result, the specification (appendix) requires that a prospective vendor endurance test a 30 bearing lot of aircraft ball bearings restored in accordance with this specification to qualify to restore ball bearings only. In addition, in order to qualify for roller bearing restoration, the specification calls for the vendor to endurance test a 30 bearing lot aircraft roller bearings. Once the vendor performs these tests, further testing will not be required. The original equipment bearing vendor is exempt from this testing only for those bearings which he has originally manufactured and subsequently restored. Should this vendor decide to restore bearings of another manufacturer, the vendor would be required to endurance test bearings to ensure the necessary life potential.

Quality control will be the responsibility of the military overhaul facility and the vendor. Current quality control procedures if properly enforced should result in quality equivalent to new bearings being purchased.

Evaluation of restored tapered-roller bearings has not been examined in this program. However, there is no reason to believe that bearing restoration by grinding, as described in this paper, could not be applied to tapered-roller bearings which are manufactured from case-carburized steels.

SUMMARY OF RESULTS

A joint program was undertaken by the NASA Lewis Research Center and the Army Aviation Systems Command to restore by grinding those rolling-element bearings which are currently being discarded at aircraft engine and transmission overhaul. Three bearing types were selected from the UH-1 helicopter engine (T-53) and transmission for the pilot program. These bearings were a 210-size split-inner-ring ball bearing and a 111-size cylindrical roller bearing used on the T-53 engine compressor shaft, and a 7216-size angular-contact ball bearing triplex set from the transmission input pinion shaft. Groups of each of these bearings were visually and dimensionally inspected for suitability for restoration. Restorable bearing yields were determined for each bearing type on the basis of bearings (or sets) and on the basis of components (rings and separators).

Fifty of each of the split-inner ring ball bearing and the cylindrical roller bearing and 50 triplex sets of the angular-contact ball bearing were restored. The restoration included regrinding raceways, plating and regrinding bores, outside diameters, and faces, stripping and replating separators, and installing new oversize balls or rollers while maintaining clearances and external geometries and tolerances in accordance to original drawing specifications.

Endurance tests were performed on 30 of each of the three types of restored bearings to verify that they will satisfactorily operate at speed, load, and lubrication conditions typical of the respective application for a period of 1600 hours. The bearings were subsequently disassembled and examined to determine the success of the restoration and endurance test program. The following results were obtained:

(1) No bearing failures occurred in the 1600 hour endurance tests with all three types of bearings which could be related to the restoration by grinding process.

(2) The bearing failures that occurred, both due to defective rolling elements, were typical of "infant" failures which may occur in new bearings.

(3) The restorable component yield (rings and separators) was 98.2 percent for the split-inner-ring ball bearing, 96.7 percent for the cylindrical roller bearing, and 91.6 percent for the triplex angular-contact ball bearing.

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APPENDIX

SPECIFICATION FOR RESTORING BEARINGS BY GRINDING

1. SCOPE

1.1 Scope. This specification presents requirements for the restoration by grinding of rolling-element bearings.

1.2 Definitions. For purposes of this specification, the following definitions shall apply:

Rolling-element bearing - A rolling-element bearing is a bearing having an inner race, an outer race, and a plurality of either balls, cylindrical rollers, crowned rollers, or spherical rollers.

Rolling-element - A rolling element is either a ball, a cylindrical roller, or a spherical roller.

Cage or separator - A cage or separator separates positions and/or retains the rolling elements between the inner and outer raceway.

Restoration - The process of grinding raceways and replacing rolling elements in accordance with this specification.

Vendor - Contractor or bidder for restoring by grinding.

Purchaser - U.S. Government or its agencies unless otherwise specified.

2. APPLICABLE DOCUMENTS

2.1 The following documents shall form a part of this specification where applicable to the extent specified herein. Unless a specific issue is specified, the latest revision shall apply.

ORIGINAL MANUFACTURER'S DRAWINGS AND SPECIFICATIONS

AEROSPACE MATERIALS SPECIFICATION

AMS 6490	Steel Bars, Forgings and Tubing 4.0 CR-4.25 Mo-1.0V (0.77-0.85C) Premium Bearing Quality, Consumable Electrode Vacuum Melted
AMS 6444D	Bars, Forging, and Mech. Tubing 1.45 Cr (0.98-1.10C) Premium Bearing Quality, Consumable Electrode Vacuum Melted

AMS 2410E	Plating-Silver, Nickel Strike, High Bake
AMS 2412E	Plating-Silver, Copper Strike, Low Bake
AMS 2406E	Plating-Chromium, Hard Deposit
AMS 2404A	Plating-Nickel, Electroless

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

ASME Life Adjustment Factors for Ball and Roller Bearings - An Engineering Design Guide, 1971.

AMERICAN NATIONAL STANDARDS INSTITUTE

ANSI B46.1 Surface Texture

AMERICAN SOCIETY FOR TESTING MATERIALS

ASTM E18	Rockwell Hardness and Rockwell Superficial Hardness of Metallic Materials
ASTM E112	Average Grain Size of Metals

MILITARY

MIL-B-197	Bearing, Rolling element, associated parts

3. REQUIREMENTS

3.1 Inspection

3.1.1 All bearings for restoring shall be inspected by vendor prior to restoring at vendor's place of business and at vendor's cost.

3.1.1.1 All inspection and test procedures shall be in accordance with ASTM methods unless otherwise agreed upon by the vendor and the purchaser. Where ASTM methods do not exist, the vendor shall submit inspection and test procedures to the purchaser for approval.

3.1.1.2 The vendor shall inform the purchaser of the test and inspection procedures to be used. Once these procedures are established they shall not be changed without prior approval in writing of the purchaser.

3.1.2 All bearing components for restoring except the rolling elements shall be visually, dimensionally, and hardness inspected in accordance with 3.1.1 hereinabove.

3.2 Rejection

3.2.1 All bearing rolling elements shall be rejected without inspection.

3.2.2 Cages or separators not meeting original manufacturer's engineering drawing dimensions or specifications or exhibiting gross wear shall be rejected. All plating shall be removed from separators prior to inspection.

3.2.3 All races having nicks, dents, or other damage on the raceway surface extending to a depth greater than 0.002 inch and/or whose hardness does not conform to original manufacturer's engineering drawing specifications shall be rejected.

3.2.4 Bearings which have been previously subjected to restoration by grinding shall be rejected.

3.3 Raw Materials

3.3.1 AISI M-50 material supplied to this specification for rolling elements shall conform to all requirements of AMS 6490 except as described in the following paragraphs.

3.3.1.1 The AISI M-50 steel supplied to this specification shall have the following composition:

Carbon	0.80-0.85	Molybdenum	4.00-4.50
Silicon	0.15-0.35	Vanadium	0.90-1.10
Phosphorous	0.10-0.25	Nickel	0.10 max.
Sulfur	0.015 max.	Cobalt	0.25 max.
Chromium	0.010 max.	Tungsten	0.25 max.

3.3.2 AISI 52100 material supplied to this specification for rolling elements shall conform to all requirements of AMS 6444D except as described in the following paragraphs.

3.3.3 Materials other than those of paragraphs 3.3.1 and 3.3.2 shall conform to applicable AMS requirements. If, however, AMS requirements do not exist for the material, the material shall conform to the original bearing manufacturer's requirements.

3.3.4 Melting Method - Material shall be produced by vacuum induction melting followed by vacuum consumable electrode remelting. The vacuum consumable electrode remelting process shall be capable of consistently producing ingot that is uniform, yielding a billet free from voids and segregation. Inclusions shall not exceed the limits of AMS 6490. The critical vacuum arc, consumable electrode remelting process variables shall be continuously monitored and recorded by the melter, and in the event that they go beyond the established limits, in an isolated instance, that area of the ingot shall not be used. The established melting practices shall not be changed without informing the purchaser and receiving approval in writing therefor.

3.4 Process Requirements

3.4.1 All raw material shall be procured only from sources approved by the purchaser.

3.4.2 Vendor may chrome or nickel plate in accordance with the requirements of AMS 2406E or AMS 2404A, the bore surfaces of the inner rings and outside diameter surfaces of the outor rings, face surfaces, and race land surfaces with an alloy to restore these surfaces to original dimensions.

3.4.3 Vendor shall grind and superfinish raceway surfaces and flange surfaces on roller bearing races removing material from said surfaces to a depth not less than 0.002 inch nor more than the depth of maximum shear stress under the maximum operational load conditions for the bearing. The vendor shall define the depth of maximum shear stress. The inner and outer raceways and flanges shall be ground to equal depths.

3.4.4. Vendor shall install the correct number of new rolling elements in the bearings.

3.4.4.1 New balls installed in the bearing shall have a diameter equal to the original ball diameter plus twice the grinding depth specified in paragraph 3.4.3. The original print IRC shall be maintained.

3.4.4.2 New rollers installed in the bearing shall have a diameter equal to the original roller diameter plus twice the grinding depth specified in paragraph 3.4.3. The roller length shall be equal to the original roller length plus twice the grinding depth specified in paragraph 3.4.3. The original print IRC shall be maintained.

3.4.4.3 Material for new balls or rollers to be installed in the bearings shall conform to the specifications in paragraphs 3.3, 3.5, 3.7, and 4.

3.4.4.4 Where specified by the original bearing manufacture, separators shall be plated in accordance with AMS 2410 or AMS 2412 as applicable.

3.4.5 Vendor shall inform the purchaser of all manufacturing processes and procedures and inspection and quality control procedures used to produce parts to this specification and receive written approval from the purchaser therefore. Once these practices are established, they shall not be changed without informing the purchaser, and receiving prior approval in writing from the purchaser therefor.

3.5 Furnished Part Requirements

3.5.1 Hardness - All bearing rollers and balls shall have an average hardness of Rockwell C62 to 64. All hardness values of these rolling elements shall be in the range Rockwell C61 to 65. Race hardness shall be according to original bearing manufacturer's specification unless otherwise specified by the purchaser.

3.5.2 Dimensions and Tolerances - All dimensions and tolerances shall be in accordance with original manufacturer's engineering drawings and specifications except as described in the following paragraphs.

3.5.2.1 All balls and roller dimensions shall be in accordance with paragraph 3.4.4.

3.5.2.2 Inner raceways may have diameters less than original specifications by an amount equal to twice the grinding depth specified in paragraph 3.4.2.3. Outer raceways may have diameters greater than original specifications by an amount equal to twice the grinding depth specified in paragraph 3.4.3.

3.5.2.3 Raceway groove radius shall be modified to reflect larger ball diameter while maintaining the same conformity as in the original specifications.

3.5.2.4 Width between flange surfaces of cylindrical roller bearings shall be increased by an amount equal to twice the grinding depth specified in paragraph 3.4.2.3 where applicable.

3.5.2.5 Internal radial and axial clearance, all other applicable internal geometry, and roller-to-race end clearance as applicable, shall conform to original bearing manufacturer's specifications.

3.5.3 Surface Finish

3.5.3.1 The surface finish of balls shall be 2 microinches AA or better; ball raceways shall be in accordance with original specifications or better.

3.5.3.2 The surface finish of rollers shall be 6 microinch AA or better on the O. D. and the ends. Roller raceways shall be in accordance with original specifications or better.

3.5.3.3 Raceway surface finish measurements shall be made in the axial direction.

3.5.4 Retained Austenite - The retained austenite content shall not exceed 3 precent as measured in accordance with paragraph 4.3.1 hereinbelow.

3.6 Traceability

3.6.1 Vendor shall assign to each restored bearing a number preceded by the letter "R" and the vendors trade mark or symbol. This number shall be marked on a face of each of the inner and outer races.

3.6.2 Records shall be maintained by the vendor to provide traceability for each new part to its corresponding heat treat lot, and heat of steel.

3.6.3 Records shall be maintained by vendor of all restored bearings by original part number, restored bearing number, serial number, and date of shipment for a period of 15 years from the date of shipment.

3.7 Material Record Requirements

3.7.1 Material identification record shall be retained by the bearing vendor for 15 years from the date of completion of the order. When requested by the purchaser, the records shall be made available for delivery within three working days. This record shall include, as a minimum, the following information:

(a) Forging vendor's certificate of test, as applicable

(b) Purchase order number

(c) Specific heat treatment cycle used (forger and/or bearing vendor)

(d) Numerical results of all required tests and inspections

(e) Macro and micro examination test results

(f) Heat number

(g) This specification number, CLASS and revision number

3.8 Qualification Testing

3.8.1 A lot of ball and roller bearings restored in accordance with this specification, where applicable, shall be tested by the vendor at no cost to purchaser to establish the reliability of bearings restored according to the vendor's manufacturing processes and procedures used to produce parts to this specification except as noted in paragraph 3.8.2 hereinbelow.

3.8.1.2 A lot of bearings for qualification testing shall comprise not less than 30 bearings.

3.8.1.3 Test conditions shall reasonably duplicate in a test rig those maximum speed, load and temperature conditions to which the bearing is subjected in aircraft applications. Bearing lubrication and lubricant shall be similar to that in the aircraft application. Test duration shall be the aircraft TBO time or the bearing L_{10} life calculated in accordance with ASME methods whichever be the shortest duration.

3.8.1.4 All bearings in a lot shall run to the time specified in paragraph 3.8.1.3 hereinabove without failure in the raceways or cage. Should any bearing or bearings fail by lubrication starvation because of test rig failure, said bearing or bearings shall be eliminated from the lot and other restored bearings substituted in their place.

3.8.2 Vendors Exempt from 3.8.1

3.8.2.1 Original vendor of proposed bearings to be restored are exempt from provisions of paragraph 3.8.1 but only to those specific bearing part numbers which said vendor manufactured.

3.8.2.2 Vendors who have restored bearings by grinding in accordance to this specification and which bearings have been tested in accordance with paragraph 3.8.1 inclusive shall be exempt from further testing providing the test results and documentation are made available to purchaser.

4. QUALITY ASSURANCE PROVISIONS

4.1 General

4.1.1 All test procedures except as herein specified shall be in accordance with ASTM methods unless otherwise agreed upon by the vendor and the purchaser in writing.

4.1.1.1 The vendor shall inform the purchaser of the test and inspection procedures to be used. Once these procedures are established they shall not be changed without prior approval of the purchaser.

4.1.2 Samples, representative of the shape and size of each forging and which have been processed through all forging operations along with the parts they represent, shall be tested to show conformance to the requirements of this specification. The frequency and the number and types of tests shall be performed to the requirements of a Quality Control Plan approved by the purchaser.

4.1.3 Finished parts shall be periodically cut-up and tested to the requirements of this specification. The frequency of testing and the number and types of tests shall be performed to the requirements of a Quality Control Plan approved by the purchaser.

4.2 Grain Size

4.2.1 Grain size shall be determined per ASTM E112 on samples representing each heat treat lot of material. For referee tests, grain size shall be determined by comparison of a polished and etched specimen with the chart in ASTM E112.

4.3 Retained Austenite

4.3.1 Retained austenite shall be determined by x-ray diffraction techniques unless otherwise approved by the purchaser. If methods other than x-ray diffraction are used, the method shall be calibrated to and show agreement with x-ray diffraction measurement.

4.4 Hardness

4.4.1 Hardness tests shall be conducted in accordance with ASTM E18.

4.5 Surface Finish

4.5.1 Surface finish shall be determined in accordance with ANSI B46.1.

5. PREPARATION FOR DELIVERY

5.1 Packing

5.1.1 All parts shall be suitably packed in accordance with MIL-B-197 to prevent damage or loss in shipment. (Class of preservation shall be specified by vendor.)

5.2 Marking

5.2.1 Each shipment shall be legibly marked, as a minimum with the purchase order number, manufacturer's name, part name, and part identification numbers.

6. NOTES

6.1 Classification of Characteristics

CRITICAL:	3.2.3, 3.4.3, 3.4.4.1, 3.4.4.3, and 3.5.1
MAJOR:	3.5.2 and 3.5.3
MINOR:	All other paragraphs

REFERENCES

1. Carter, T. L., " A Study of Some Factors Affecting Rolling-Contact Fatigue Life," NASA TR R-60, 1960.

2. Bear, H. R., and Butler, R. H., "Preliminary Metallographic Studies of Ball Fatigue Under Rolling-Contact Conditions," NACA TN-3925, 1957.

3. Carter, T. L., Butler, R. H., Bear, H. R., and Anderson, W. J., "Investigation of Factors Governing Fatigue Life with the Rolling-Contact Fatigue Spin Rig," ASLE Trans., Vol. 1, No. 1, Apr. 1958, pp. 23-32.

4. Jones, A. B., "Metallographic Observations of Ball Bearing Fatigue Phenomena," Symposium on Testing of Bearings, ASTM, 1947, pp. 35-52.

5. Parker, R. J., "Correlation of Magnetic Perturbation Inspection Data with Rolling-Element Bearing Fatigue Results," ASME Trans., Jour. Lub. Tech., Vol. 97 No. 2, 1975, pp. 151-158.

6. Chevalier, J. L., and Zaretsky, E. V., "Effect of Carbide Size, Area, and Density on Rolling-Element Fatigue," NASA TN D-6835, 1972.

7. Chevalier, J. L., Zaretsky, E. V., and Parker, R. J., "A New Criterion for Predicting Rolling-Element Fatigue Lives of Through-Hardened Steels," ASME Trans. Jour. Lub. Tech., Vol. 95, No. 3, 1973, pp. 287-297.

8. Cobb, L. D., "Effect of Vacuum Melting on Bearing Steel," Paper presented at the SAE National Passenger Car, Body, and Materials Meeting, Detroit, Mich., Mar. 5-7, 1957.

9. Johnson, R. F., and Sewell, J. F., "The Bearing Properties of 1 Percent C-Cr Steel as Influenced by Steelmaking Practice," Jour. Iron and Steel Institute, Vol. 196, Pt. 4, Dec. 1960, pp. 414-444.

10. Morrison, T. W., Tallian, T., Wolf, H. D., and Baile, G. H., "The Effect of Material Variables on the Fatigue Life of AISI 52100 Steel Ball Bearings," ASLE Trans., Vol. 5, No. 2, 1962, pp. 347-364.

11. Schlatter, R., and Stromp, J. P., "Improved M-50 Aircraft Bearing Steel Through Advanced Vacuum Melting Processes," J. Vac. Sci. Technol., Vol. 9, No. 6, 1972, pp. 1326-1333.

12. Bamberger, E. N., Zaretsky, E. V., and Signer, H., "Endurance and Failure Characteristics of Mainshaft Jet Engine Bearings at 3×10^6 DN," ASME Paper no. 76-LubS-16, 1976.

13. Martin, J. A., and Eberhardt, A. D., "Identification of Potential Failure Nuclei in Rolling-Contact Fatigue," ASME Trans., Jour. Basic Engr., Vol. 89, No. 4, 1967.

14. Littman, W. E., and Widner, R. L., "Propagation of Contact Fatigue from Surface and Subsurface Origins," ASME Trans., Jour. Basic. Engr., Vol. 88, No. 3, 1966, pp. 624-636.

15. Leonard L., Martin, J., and Choman, L., "Special Report on Surface and Subsurface Observations of Endurance Tested 6309-Size Bearings," SKF Report AL69M025, Oct. 1969.

16. Stanley, D. C., "Bearing Field Inspection and Refurbishing," Paper presented at Symposium on Propulsion System Structural Integration and Engine Integrity, Naval Post Graduate School, Monterey, Calif., Sept. 3-6, 1974.

17. Bamberger, E. N., Zaretsky, E. V., and Anderson, W. J., "Effect of Three Advanced Lubricants on High-Temperature Bearing Life," ASME Trans., Jour. Lub. Tech., Vol. 92, no. 1, 1970, pp. 23-33.

18. Li, D. F., Kauzlarich, J. J., and Jamison, W. E., "Surface Roughness Effects on Fatigue in Partial EHD Lubrication," ASME paper no. 76-LubS-2, 1976.

TABLE I. - BEARING IDENTIFICATION AND SPECIFICATION

Bearing	210-Size split-inner ring ball bearing	111-Size cylindrical roller bearing	7216-Size angular-contact ball bearing
Federal stock number	3110-727-3032	3110-071-4568	3110-135-2603
Component part number	1-300-015-04	1-300-176-03 (1-300-176-04)	205-040-246-3
Tolerance	ABEC-5	RBEC-5	ABEC-5
Contact angle	27° to 30°	---------------	25°
Bearing steel	AISI 52100	AISI M-50	AISI M-50
Number of rolling elements	14	16 (20)	15
Cage material	Bronze	Steel (bronze)	Steel
Cage type	One piece, inner-land riding	One piece, inner-land riding	One piece, inner-land riding

TABLE II. - RESTORABLE BEARING YIELD

	210-Size split-inner ring ball bearing	111-Size cylindrical roller bearing	7216-Size angular contact ball bearing triplex sets
Number of bearings (or sets) for inspection	150	121	86
Number of bearings (or sets) restorable	145	114	55
Restorable bearing yield, percent	96.7	94.2	64.0
Number of restorable components	442	351	701*
Restorable component yield, percent	98.2	96.7	91.6

*The 86 triplex sets included 765 components, since three bearings from two sets were not returned from the overhaul depot for inspection.

TABLE III. - ENDURANCE TEST CONDITIONS

Bearing	210-Size split-inner ring ball bearing	111-Size cylindrical roller bearing	7216-Size angular-contact ball bearing
Spindle speed, rpm	24 000	24 000	6600
Radial load, lb	0	100	1905
Thrust load, lb	450	0	2564
Lubricant inlet temperature, $^{\circ}F$	195±5	195±5	150±10
Lubricant flow rate, gpm	0.38±0.043	0.38±0.043	2.1±0.3
Sump capacity, gal	4	4	12
Number bearings per sump	4	4	8
Lubricant change interval, hr	100	100	600

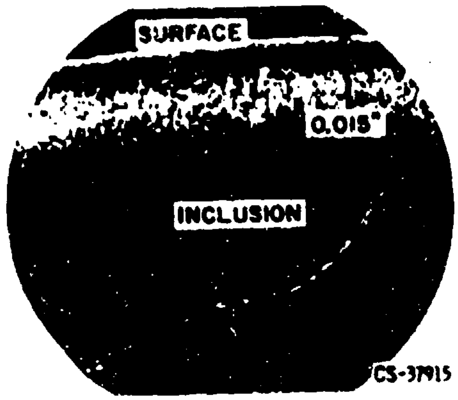

Figure 1. - Fatigue crack emanating from an inclusion.

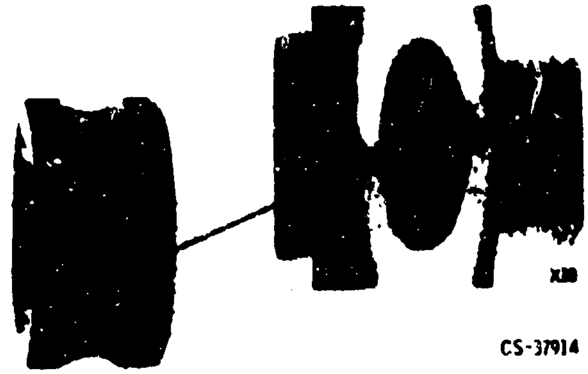

Figure 2. - Typical fatigue spall in bearing race.

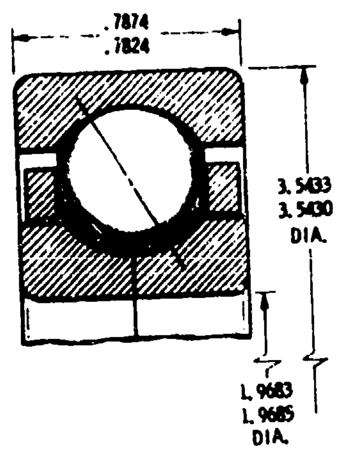

Figure 3. - 210-size split inner-ring ball bearing.

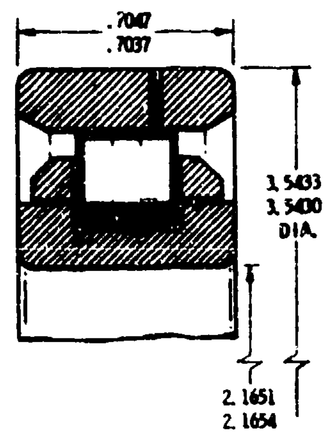

Figure 4. - 111-size cylindrical roller bearing.

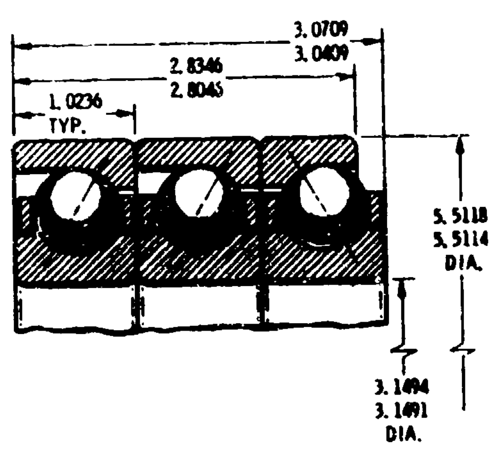

Figure 5. - 7216-size angular-contact ball bearing triplex set.

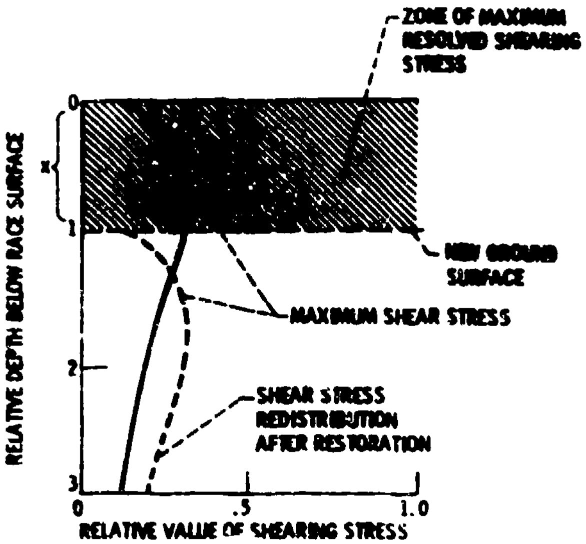

Figure 6. - Relative value of shearing stress as function of depth below rolling-element surface.

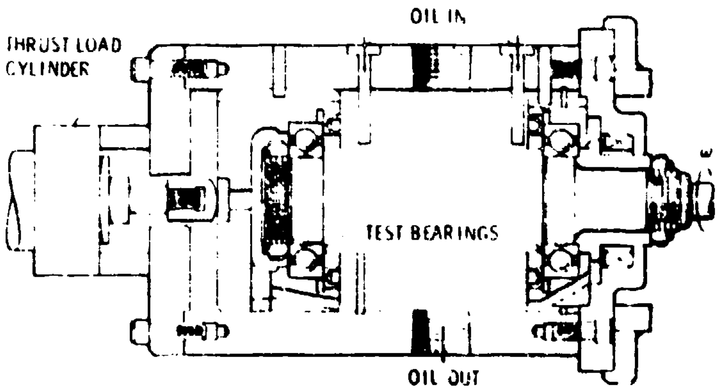

Figure 7. Test head for 210-size split-inner ring ball bearings.

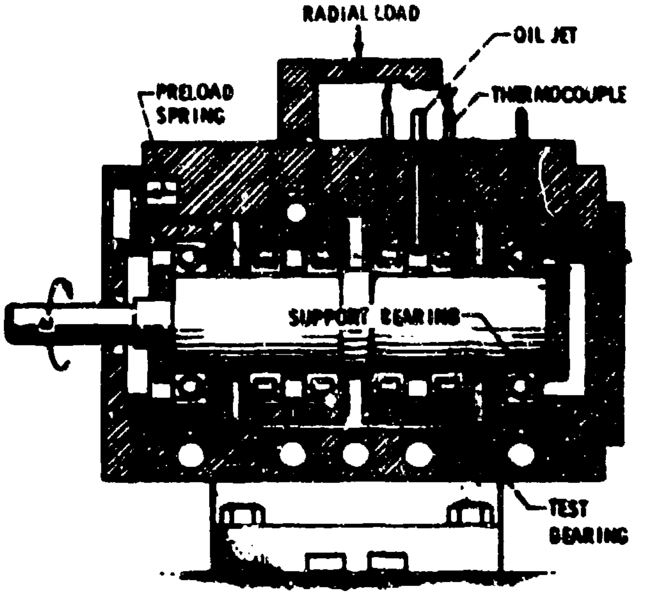

Figure 8. - Test head for 111-size cylindrical roller bearings.

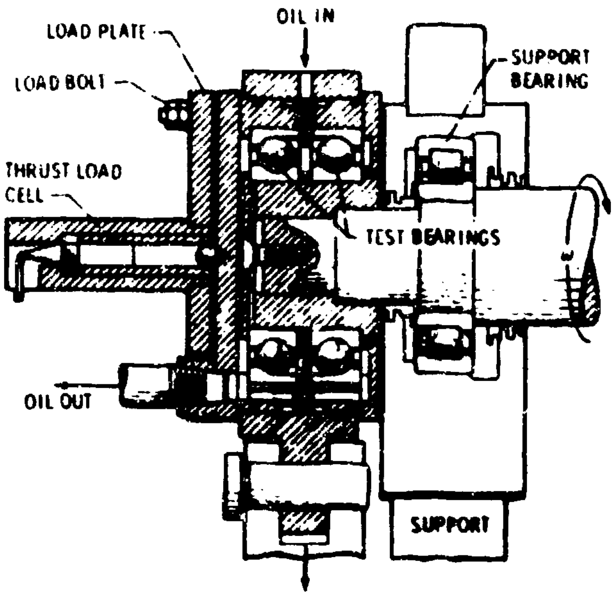

Figure 9. Test head for 7216-size angular contact ball bearing.

NASA-Lewis

ROLLING-ELEMENT BEARING RESTORATION - A USERS VIEWPOINT

by Hubert L. Bull

Corpus Christi Army Depot

ABSTRACT

Ball and roller bearings restored by grinding were inspected and tested in T-53 engines and UH-1 transmissions in test cells at Corpus Christi Army Depot (CCAD). Inspection of the restored bearings including nondestructive magnetic testing and visual, laser scattered light and scanning electron microscope inspections revealed that the restored bearing surfaces were comparable to new bearings. Tests in three engines and three transmissions to time periods up to 150 and 1500 hours, respectively, revealed no wear or abnormalities in the restored bearings. All bearings from these tests were in excellent condition. Implementation of a restoration by grinding program for the Army at CCAD is recommended.

INTRODUCTION

A major concern facing Army aviation is the replacement cost of component parts such as rolling-element bearings. Rolling-element bearings are designed for an engine or transmission application based upon the application design load and the life and reliability expected to assure long term operation. Life for these bearings are based upon a rolling-element fatigue criteria (ref. 1). However, only 10 percent of the bearings that are removed have failed from fatigue and are not reusable. The other bearings which are scrapped generally have surface damage of the raceways or the dimensions are not within print. These bearings can account for as much as 50 percent of the bearings removed at overhaul. In most cases the potential remaining fatigue life of these bearings are many times the accumulated hours already on the bearing. Current refurbishing methods unfortunately cannot save these bearings and at the same time maintain the reliability required for aviation applications.

The NASA Lewis Research Center suggested to the Army Aviation Systems Command and the Corpus Christi Army Depot a means to salvage those bearings which are currently being scrapped by grinding the raceways. This method was based upon basic research performed by the NASA and others (ref. 2).

The cost implications of such a program was appealing. In addition, the delivery time for obtaining restored bearings would be expected to be from one-third to one-half that for obtaining new bearings. This would greatly assist CCAD logistics.

A pilot program to restore bearings by grinding was initiated by the Army Aviation Systems Command (AVSCOM) in conjunction with CCAD, the NASA Lewis Research Center, and the Industrial Tectonics, Inc. (ITT) Bearing Division. The NASA Lewis Research Center was to furnish to the Army the engineering and technical coordination required to implement the program; AVSCOM was to provide project management and endurance testing through private industry; ITI was to undertake the bearing restoration by grinding in conjunction with CCAD; and CCAD was to furnish candidate bearings, post restoration inspection and qualification testing on engines and transmissions. This paper describes the work undertaken by CCAD and the results thereof.

PILOT PROGRAM

In order to establish the technical and economic feasibility of rolling-element bearing restoration by grinding a contract was awarded Industrial Tectonics, Inc. to examine 529 bearings removed at overhaul from UH-1 helicopter transmissions and engines. These bearings which are described in detail in reference 2 comprise the 210-size split-inner-ring ball bearing and the 111-size cylindrical roller bearing which are the compressor front and rear bearings, respectively, of the T-53 engine, and the 7216-size angular-contact ball bearing triplex set from the transmission input pinion shaft.

Thirty sets of each bearing type which were restored by grinding were sent to the NASA for testing under AVSCOM contract under simulated field conditions for a time of 1300 hours. This time is the engine and transmission TBO (ref. 2).

While a great deal of statistical confidence can be obtained from bench type bearing endurance testing, these tests cannot completely simulate or duplicate the actual engine or transmission dynamic and thermal conditions. Therefore, it was considered important that a supplemental and complementary program be undertaken by CCAD to test representative bearings in actual engine and transmissions to assure a greater degree of confidence. Twenty sets of each group of restored bearings were sent to CCAD. These bearings were from the same production lots as those sent to the NASA for endurance testing.

Prior to testing the bearings in the respective engines and transmissions and after visual inspection by CCAD, the three groups comprising 20 bearing

sets were sent to the Southwest Research Institute (SWRI). At SWRI the bearings were subjected to advanced nondestructive testing techniques, including magnetic perturbation, and Barkhausen noise. The bearing surfaces were also inspected by a laser scattered light technique.

The magnetic perturbation technique was used to nondestructively detect surface and subsurface flaws (inclusions, cracks, pits, or dents) in the bearing raceways. The method consists of establishing a magnetic flux in the region of the material being inspected, and then scanning the surface of the material with a sensitive magnetic probe. The probe detects anomalies or perturbations in the magnetic flux which are due to the presence of flaws.

The Barkhausen noise method was used to nondestructively detect residual stresses in the raceway subsurface. This technique uses a probe to detect Barkhausen jumps, or abrupt movements of the boundaries of magnetic domains, as the part is magnetized. The presence of residual stresses in the raceway influences these movements and thus the signal received from the probe.

These bearings were subsequently sent to SKF Industries, Inc. for a characterization of the bearing surfaces by visual observations under magnification (×50) and photographic documentation utilizing scanning electron microscopy.

Generally, four characteristic classification of surface irregularities are prominent on the restored bearing raceways.

1. Surface disturbances
2. Scratches
3. Finishing damage
4. Pits

The term surface disturbances is used to represent any one of several surface defects which are generally quite small in nature and result in a disruption in the as-ground surface.

Other potential defects that were apparent were from removal or disruption of the black oxide coatings on the 7216-size angular-contact ball bearings which were black oxide coated prior to and subsequent to restoration.

A general summary of the results from the various inspection techniques for each of the bearing types is presented in Table 1. Inspection of the bearing raceways revealed that with few exceptions the bearings were comparable to those of new bearings. While a criteria for bearing rejection has not been established, six each of the 210-size split-inner-ring ball bearing and the 111-size cylindrical roller bearing which exhibited the largest and most serious number of defects were chosen for insertion and testing in the T-53 engine. Three triplex sets of the 7216-size angular-contact ball bearing were selected for testing in the UH-1 helicopter transmission.

ENGINE AND TRANSMISSION TESTING

UH-1 Helicopter Transmission

The transmission in the UH-1 helicopter is rated at approximately 1400 SHP from the engine. Input speed is 6600 rpm with an output speed at the rotor of 324 rpm. The oil-in temperature in the transmission is approximately 140° to 160° F with an oil-out temperature of approximately 200° F.

The transmission is located directly ahead of the engine and is suspended by Pylon isolating mounts on structural support. The unit is coupled to the engine through a short drive shaft. It provides drive angle change and speed reductions, through a train of spiral bevel gears and a two stage planetary to drive the main rotor mast. A free wheel clutch in the input drive quill coupling disengages to allow main rotor and gear train to turn freely when the engine is stopped or is idling below rotor driving speed, as in auto-rotational descend.

The triplex set of 7216-size angular-contact ball bearings are mounted on the input bevel gear pinion shaft. The bearings rotate at a speed of 6600 rpm. The set is loaded with a 3175-pound radial and a 4274-thrust load at 1250 HP take-off power. The first two bearings in the triplex set are preloaded with a 50 pound axial load. The third bearing on the set is match ground with a 470 pound axial preload.

T-53 Engine

The T-53 gas turbine engine is a free turbine type power plant developed for use in rotary wing aircraft. The engine consists of an inlet section, compressor section, diffuser section, combustion and exhaust section. All sections are designed so as to include an annular flow path for the air or hot gases. The sections are structurally interdependent. They support all internal rotating systems and provide attaching capabilities for engine required external components and limited airframe accessories.

The engine is rated at 1400 SHP. The compressor operates at 24 000 rpm and the power turbine at 21 000 rpm. The power turbine speed is reduced through a 3.2:1 gear box to a speed of 6600 rpm.

The 210-size split-inner-ring ball bearing is the front compressor shaft bearing. The load on the bearing is approximately 450 pound axial load. The oil-in temperature is approximately 200° F with an oil-out temperature of 250° to 300° F.

The 111-size cylindrical roller bearing is located on the compressor rear shaft. The radial loads on the bearing are nominal and mainly due to unbalances

due to aerodynamic loads. The oil-in temperature is approximately 200° F and the oil-out temperature is approximately 350° F.

RESULTS AND DISCUSSION

Engine Tests

Two restored 210-size split-inner ring ball bearings and two restored 111-size cylindrical roller bearings were tested in a T53-L13B engine. The bearing positions were as follows:

Engine position number	Bearing designation
1	Ball-210 size
2	Roller-111 size
3	Roller-111 size
4	Ball-210 size

This engine had several test items installed, but none were integral to the oil system and failure would not have affected the bearings. The engine was installed in CCAD Test Cell No. 5 and a standard production run performed 4 April 1975. The engine produced military rated power at 100.4 percent N1 rpm at 1105° F exhaust gas temperature referred to standard day conditions. The bearing scavenger temperature of the No. 2 bearing package was 325° F and the scavenger from the No. 4 bearing was 337° F with an oil inlet temperature of 196° F and a 65° F ambient temperature at military rated engine power.

The actual test began on 7 April 1975 and was run on a two-shift basis until completion on 21 April 1975. The qualification test was conducted in accordance with Lycoming Test Spec XTS313.4.1B. This test requires 150 hours of test which accumulates an actual run time of 178 hours. The 178 hours includes a standard production run before and after the test.

A spectrographic oil analysis was taken each 6-hour test cycle. There were no unusual metal contamination during the test and no unusual problems otherwise during the test.

The initial vibration level of the engine was well within the specifications. The vibration after-test during the final production run was well within specified limits. The coastdown time of the first production run was 61.0 seconds and on the final production run was 63.9 seconds. The scavenger oil temperatures at the end of the test were the same as the beginning production run with oil inlet at 194° F and an ambient temperature of 82° F at military rated power.

A second engine test was run with four more restored bearings with a second T53-L13B engine. This engine produced military rated power at 99.7 percent N1 rpm with a 1052^{O} F exhaust gas temperature referred to standard day conditions. The oil temperature and vibration readings were within specifications at the beginning and end of test run. This test was also conducted in accordance with Lycoming Test Spec XTS313.4.1B and no unusual problems were noted.

A third engine was run in Test Cell No. 4 with four more restored bearings. The beginning production run was made 21 May 1975. The engine produced military rated power at 100.1 percent N1 rpm and 1044^{O} F exhaust gas temperature at standard day conditions. The oil-out temperature for the roller bearings was 370^{O} F and for the power turbine bearings 342^{O} F. The oil inlet temperature was 198^{O} F at an ambient temperature of 77^{O} F.

The engine was removed from the test cell and reinstalled in Test Cell No. 5 on 2 June 1975. The qualification test was conducted in accordance with Lycoming Test Spec XTS313.4.1B. This test requires 150 hours of test time which came out 178:45 total engine running time. This includes a before- and after-standard production test and 161 engine start-and-stop cycles. A spectrographic oil analysis was run by the laboratory with a sample being taken each day. A vibration at flight idle (F/I) increased until it was decided to discontinue the test after 74 hours.

The engine oil filter and the test cell filter were removed. The filters contained metal traces. The oil sample also had a high iron content.

The engine was disassembled and the bearings inspected with no discrepancies found. The bearings showed no signs of damage. The gear box and the reduction gears were inspected with no signs of failure. The engine was reassembled and the test resumed. The high vibration problem was corrected, but as the test progressed, the vibration level at F/I increased until the completion of the test when it was near the maximum limits allowed. Inspection of the bearings after-test revealed no wear or damage.

The testing was performed on a two-shift basis until completed on 15 July 1975. No other unusual problems occurred during the test. The only discrepancies noted were an oil leak on the starter drive shaft and the fuel control drive shaft.

The oil-out temperature was 336^{O} F for the roller bearings and 303^{O} F for the turbine bearings. The oil inlet temperature was 191^{O} F at an ambient temperature of 86^{O} F. The coastdown time at the beginning of the test was 63.0 seconds and 84.3 seconds after completion of the test. In all tests reported the bearings were in excellent condition.

Transmission Tests

Transmission tests with the restored triplex bearing sets began in April 1975. The first test was conducted simulating as nearly as possible, a flight profile of the Lycoming Engine Test Spec XTS313.4.1B in 6-hour cycles. Total time accumulated in this test was 300 hours with no abnormalities noted.

A second transmission test was run with a second set of restored bearings installed. This test was for 300 hours total time and was tested to the same specifications as the previous transmission. No abnormalities were noted.

A third transmission was tested with a new set of restored bearings installed. This test was run 1500 hours total time without inspection until completed. Inspection revealed no excessive wear and no abnormalities were noted during test.

GENERAL COMMENTS

The restored bearings selected for these tests were those with the largest number of defects and more serious indications from the baseline inspection data. It was concluded from these tests that the restored bearings performed in a highly satisfactorily manner. It is recommended that the technique of restoration by grinding be qualified for use by the Army to recover assets which would otherwise be scrapped. It is further recommended that the program of restoration by grinding be implemented at CCAD. A specification for restoration by grinding of Army aircraft bearings should be developed and adopted by the Army. It is further recommended that the remaining bearings be installed in engines and testing be continued at the Army Test Board, Fort Rucker, Alabama.

SUMMARY OF RESULTS

A program to inspect bearings restored by grinding and, subsequently, to test the restored bearings in actual engines and transmissions was performed. Inspections included nondestructive magnetic testing and visual, laser scattered light, and scanning electron microscope inspections. Engine and transmission testing with the restored bearings were performed in CCAD test cells using simulated flight profiles with the T53-L13B engine and the UH-1 main transmission. Three engine tests were performed with a different set of restored bearings on the compressor shaft for each test. Three transmission tests were performed with a different set of restored bearings installed on the input pinion shaft for each test. All bearings were disassembled and inspected at the end of the engine and transmission tests. The following results were obtained:

1. Pretest inspection of 20 sets of bearings restored by grinding indicated that the bearing raceways were comparable to new bearings.

2. Inspection of the restored 210-size ball bearings and the 111-size roller bearings tested in three engine tests for time periods in excess of 150 hours each, revealed excellent condition of the bearing surfaces.

3. Inspection of the restored 7216-size ball bearings tested in three transmission tests to time periods to 1500 hours revealed no excessive wear and no abnormalities of the bearings.

REFERENCES

1. Littmann, W. E., "The Mechanism of Contact Fatigue," Proceedings of Symposium on Interdisciplinary Approach to the Lubrication of Concentrated Contacts, NASA SP-237, 1970, pp. 309-377.

2. Parker, R. J., Zaretsky, E. V., and Chen, S., "Evaluation of Ball and Roller Bearings Restored by Grinding," NASA TM to be published.

TABLE 1. - SUMMARY OF INSPECTION OF RESTORED BEARINGS BY GRINDING

Bearing type and number inspected	Bearings with probable irregularities as determined by various inspection methods, percent				
	Magnetic perturbation	Barkhausen noise	Laser scattered light	×50 Microscope	CCAD visual
210-Size split-inner-ring ball bearing (20 bearings)	55	75	95	100	55
111-Size cylindrical roller bearing (22 bearings)	55	45	82	95	18
7216-Size angular-contact ball bearing (60 bearings; 20 triplex sets)	28	38	77	92	0

E-8728d

MICROECONOMIC ANALYSIS OF MILITARY AIRCRAFT BEARING RESTORATION

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ABSTRACT

The risk and cost of a bearing restoration by grinding program was analyzed. A microeconomic impact analysis was performed. The annual cost savings to U.S. Army aviation is approximately $950,000.00 for three engines and three transmissions. The capital value over an indefinite life is approximately ten million dollars. The annual cost savings for U.S. Air Force engines is approximately $313,000.00 with a capital value of approximately 3.1 million dollars. The program will result in the government obtaining bearings at lower costs at equivalent reliability. The bearing industry can recover lost profits during a period of reduced demand and higher costs.

INTRODUCTION

Roller and ball bearing fatigue failures account for approximately ten percent of all bearing failures. The remaining 90 percent may be attributed to a variety of causes related to manufacturing flaws, human error or the effects of dirt and corrosion. Unlike fatigue failures, the other causes of failure are not subject to a tractable analytical procedure for failure prediction (ref. 1).

Those bearings removed for reasons other than fatigue failure consist primarily of component failures which affect the integrity of the bearing and consequently the integrity of the aircraft. In order to minimize risk, a subjective judgment is made to remove the bearing

and replace it. The usual procedure is to remove bearings during overhaul or other work on engines or transmissions, clean and visually inspect them for defects and dimensional conformance to print. Since most bearings are not disassembled during this operation, if there is any doubt regarding integrity, the bearing is replaced.

A pilot program was undertaken by the AVSCOM and NASA in conjunction with Industrial Tectonics, Inc. to establish the restorable yield of bearings which would be candidates for restoration by grinding (refs. 1 and 2). Using statistics obtained from the Corpus Christi Army Depot for a three month period in 1972 for the UH-1 helicopter, out of 4212 rolling-element bearings which were removed and discarded at overhaul for a calendar quarter it was probable that 90 percent or 3792 could be recovered through restoration by grinding. Extrapolating these statistics, it was further speculated that approximately one-half million dollars a year could be saved by Army aviation through bearing restoration by grinding (ref. 2). Assuming for purposes of discussion that a 90 percent recovery can be accomplished, the risk and cost of such a procedure must be analyzed.

In addition to the above, consideration should be given to the microeconomic impact of bearing restoration upon the bearing market. In the event that there is little impact upon the market, and little or no risk of using restored bearings, then a bearing restoration program would offer the government a significant cost savings over replacement of new bearings. If any of the areas produce adverse consequences, restoration still cannot be rejected, but the analysis to determine overall value becomes considerably more complicated. In this paper, an attempt will be made to determine overall value, risks and costs associated with a bearing restoration program.

PROCEDURE

The bearing restoration procedure is nearly the same procedure as the manufacturing of new bearings except that much of the work has already been performed. The process constitutes approximately the

last 30 percent of the total operations required to make a new bearing. Of considerable significance is the fact that this portion of the process is the least capital intensive since most of the forming has been done and material replacement is confined to ball or roller elements. One manufacturer's process (ref. 2) consists of a grinding operation on the outer diameter, bore, land, face, and race. Bores and OD's are ground and either thin nickel or chrome plated to restore them to original print dimensions or salvage dimensions. Inner and outer raceways are reground to a depth of 0.002 in. per surface. Since the race radii are each approximately 0.002 in. larger, the bearing must be refitted with balls 0.004 in. larger in diameter (ref. 2). The effective race curvature is identical to original dimensions within significant mathematical values. Although the bearing contains oversized balls and raceway curvatures, the geometry of the bearing is unchanged and so the stress level and estimated life will be identical to the original bearing. A similar restoration procedure applies to roller bearings (ref. 2).

The restored bearing contains new rolling elements such as balls or rollers. An advantage of regrinding the raceways is that any incipient spalls or cracks will be uncovered; in such cases the part will be scrapped.

The final aspect of the procedure is that it requires an economic lot size production run because of the set-up cost which is about $500. Although some manufacturers consider the minimum economic lot size to be about 500 bearings, the set up cost per bearing will go from $1 to $5 if only 100 bearings are restored in a single lot. For $50 OEM bearings, the $5 per bearing set-up cost may be enough to render restoration uneconomical, but for a $900 OEM bearing, the charge may not be significant. At this point an assumption will be made: it is assumed that each bearing federal stock number will constitute one potential production lot. That is, the geometry and other characteristics of a bearing are uniquely specified by a federal stock number. As a result, it is not possible to combine various bearing types to produce an economic lot size.

In order to conduct a restoration program, it would be necessary to establish several inventory policy changes. Additionally, it would be necessary to conduct annual analyses and reviews to ensure that bearing types selected for the program are being used in sufficient quantity to maintain qualification for the program. Further, consideration should be continually given to qualify other bearing types should price or quantity changes warrant their inclusion in the program. The review and analysis should be conducted by a central authority where complete records of bearing consumption are available.

The review and analysis ideally should be a two stage process; the first stage would consist of a rough screening review and the second, a detailed cost review.

The rough screening review should consist of two criteria; these being price and quantity. The simplest criteria are: if the bearing cost exceeds $50 OEM price and annual replacement is greater than 100 bearings, the bearing passes the rough screening to become a candidate for a detailed analysis. Another useful criterion would be: if the demand-price product exceeds $5000 per year, the bearing should be a candidate for a detailed analysis.

For the bearings passing the rough screening, the detailed analysis would require the following information for each bearing:

Federal Stock Number

Annual requirements for replacement

Technical Data

- Bearing Drawing
- Use
- Restoration tolerances

This information should then be sent to potential vendors with a request for quotation of prices which should include

Unit price for 60 percent return

Unit price for 70 percent return

Unit price for 80 percent return

Unit price for 90 percent return

The RFQ should stipulate that the quotes include a guarantee that the manufacturer will restore a specified fraction of the rejected bearings or incur a penalty. An example of such a penalty where the fraction restored is less than the guaranteed return is,

(1.25)×(Guaranteed fraction return minus actual fraction return)×
(number of bearings shipped to vendor)×(Prevailing scrap price)

In this case, the vendor is required to pay scrap value plus 25 percent for the difference between his guaranteed rate of restoration and the vendor's actual performance. Also, the vendor will be required to pay the prevailing scrap price on (1 - Guaranteed Fraction) of the bearings shipped to the vendor. Moreover, the unit price per bearing paid by the government would be equal to the lowest unit price of the quoted guaranteed returns. After quotations are returned, the optimal replacement policy can be determined by selecting the minimum cost for each bearing type using the expression

For a given i

$$\mathrm{MIN}\ TC(i, j, GF_{ijk})$$

such that

$$\begin{aligned} TC(i, j, GF_{ijk}) = N_i \Big[&(GF_{ijk} \times UP_{ijk}) - (1 - GF_{ijk})(SV_i) \\ &+ (1 + GF_{ijk})(SC_{ij}) + (1 - GF_{ijk})(OEMP_i) \\ &+ (1 - GF_{ijk})(OEMPSC_i) \Big] \end{aligned}$$

where

i	bearing type
j	vendor
k	guaranteed fraction index (0.6, 0.7, 0.8, 0.9)
$TC(i, j, GF_{ijk})$	total cost of replacement for bearing type i, using vendor j with a guaranteed return rate k

N_i number of bearings of type i to be shipped to vendor

GF_{ijk} guaranteed fraction return of type i from vendor j with return index k ($k = GF_{ijk}$)

UP_{ijk} quoted price of restoring one unit of i from vendor j with guaranteed fraction k

SV_i prevailing salvage value of bearing type i

SC_{ij} one way shipping cost

$OEMP_i$ original equipment vendor unit price for bearing type i

$OEMPSC_i$ shipping cost for one new bearing type i

This optimization procedure will yield the minimum cost policy for each bearing type. A sensitivity analysis should also be performed by varying the fraction returned and including the penalty function to determine the sensitivity of not meeting the vendor expected performance. In the event that the vendor exceeds promised performance, there should be no extra incentive because the vendor will be benefiting directly since the vendor is paid for each unit and the government will benefit from a lower than expected total cost.

COST EFFECTIVENESS ANALYSIS

A partial listing of annual bearing demand was obtained from the Army Aviation Systems Command. This listing included current prices and annual demand for the following

ENGINE	TRANSMISSION
T53	UH-1
T55	CH-47
T63	OH-58

Data on certain prices in effect in 1972 were also available. Table I is a comparison of these costs. Although the sample is very small and probably not statistically adequate, the information provided is worthy of

special note. During the period 1972 to 1976, the U.S. economy was subjected to a rather severe case of demand-pull inflation. The results of such inflation are that general price levels rise because the demand for goods and services exceeds the supply available at existing prices (ref. 3). Another consequence is that there are long delays for industrial and commercial deliveries and large backlogs in manufacturing plants. The backlogs may still be present in the bearing industry for some bearing types, but the large price increases are conspicuous by their absence. Reference will be made to this point again.

An analysis was conducted using the annual demand data from AVSCOM. The data are only for the three Army engines and transmissions cited above. Tables 2 and 3 show the stock numbers, annual demand, OEM price and estimated restoration price for all bearings in the data set which pass the rough screen test criterion of demand - OEM price product > $5000 per year. Table 4 is a comparison of cost differences between OEM replacement and a 0.90 fraction (90 percent) programs. The total cost savings of the 0.90 restoration program is approximately $950,000 for one year for the three engines and three transmissions in the data set. If OEM demand cost for bearings is relatively stable despite fluctuations in actual demand for specific types, then a relatively constant savings can be accomplished through bearing restoration. This value will be approximately one million dollars per year. The capital value over an indefinite life of such a program is approximately ten million dollars for the three engines and three transmissions included in this analysis.

A similar analysis was performed using data supplied by the U.S. Air Force MATP in Oklahoma City. These data appear in tables 5 and 6. The annual cost savings of bearing restoration for the Air Force engines would amount to approximately $313,000.00. The capital value over an indefinite life of such a program is approximately 3.1 million dollars.

RISK AND UNCERTAINTY

In the previous section, cost quotes by a single bearing manufacturer were used to compare the cost-effectiveness of restoration with the purchase of new bearings. The prices are sufficiently conservative whereby it would not be generally expected that these prices would be exceeded in a competetive procurement. Therefore, the cost savings associated with the price quotations would generally be more than may be actually achieved.

The major risk and uncertainty associated with the process is the reliability of restored bearings and the yield of production runs. It was assumed that a 90 percent yield could be achieved. Therefore, the cost-effectiveness analysis was performed using a 0.90 fractional yield. This basic assumption is supported by the inspection of 529 bearings comprising three bearing types from the UH-1 helicopter (refs. 1 and 2). The inspection results indicated a potential yield rate which exceeded 90 percent which supports the assumption used herein. Additional inspection data should be obtained in order to establish the confidence of the yield rate for other bearings and bearing applications.

Based upon the endurance testing reported in reference 1 for the three bearing types discussed above, the reliability of the restored bearings appears to be very similar to that of new bearings. On the basis of these results the risk and uncertainty associated with the reliability of restored bearings are no greater than those associated with new bearings. Further testing of restored bearings from other vendors should be conducted to establish the confidence of this conclusion for other restoration by grinding methods.

If one examines the failure rate as a function of time, experience should be similar to the graph in figure 1. The population will initially exhibit a high failure rate if it contains some proportion of substandard, weak specimens (ref. 4). As these components fail, the failure rate decreases rapidly during the burn-in or debugging period and stabilizes to an approximately constant value. During the useful life period, the failure rate is at its lowest level. When the components reach the life T_W,

wearout becomes noticeable and from that time, the failure rate increases rapidly. The work of reference 1 indicates that such a "bathtub function" may be applicable to restored bearings.

It is noteworthy that the only two failures which occurred out of the 90 bearings tested were attributed to the new rolling elements. These would have also failed in new bearings. The reliability of the bearing itself may be greater than that of new bearings because of the probable elimination of infant mortality of the restored bearing raceways. Such a tentative conclusion should be reinforced with additional testing of other restored bearings.

MICROECONOMICS OF BEARING INDUSTRY

The effect of implementing a bearing restoration ogram will primarily be a doubling of the useful life of approximately 90 percent of engine and transmission bearings in the program. An obvious question is what effect such a program will have upon the bearing industry. To answer such a question, it is necessary to delve into the microeconomics of the bearing industry. A key to answering the question lies in the behavior of prices over the period of the last several years.

From 1972 to 1976, the economics of the west were plagued with double digit demand-pull inflation. Increases in aggregate demand caused rapid price increases as the industrial sector approached maximum production levels. However, as noted previously in the small sample comparison, prices of bearings increased an average 2 percent annually. Several factors could account for such anomolous stability. If demand for bearings was lower than capacity, this situation might possibly dampen price increases because of competition and because there would be li .e need for capital expansion.

During this period (1972-1970) there was a gradual withdrawal from Vietnam. Therefore a decreased demand for bearings is plausible. Over a period of a couple of years, if costs were stable and demand decreased, there would be increased competition and a tendency to reduce profits for the sake of o -imizing production costs. Such a conclusion

results from an examination of total production costs as a function of production rate. The elements of total cost are the total fixed costs of a facility and the total variable costs. Fixed costs are incurred from expenses for depreciation, taxes, utilities, and other indirect expenses. Variable costs are incurred from labor and materials. Total costs (TC), total variable costs (TVC) and total fixed costs (TFC) are depicted as a function of production rate (units per time unit) in figure 2.

In addition to these costs there is average cost (AC) which is total cost divided by the production rate. Average variable cost (AVC) is the total variable cost divided by the production rate; average fixed cost (AFC) is the total fixed cost divided by the production rate. If the rate of production is increased and fixed cost is a constant, then the AFC will decrease. The TVC has a tendency to increase rapidly as production is started, but will tend to level off to some extent as the optimal plant capacity is approached. When this point is exceeded, it will be necessary to add labor in the form of overtime or additional personnel and material costs will have a tendency to increase. The net effect is that there will be a point of inflection in the TVC function. Up to this point, AFC and AVC have been decreasing. Thus, the marginal cost (MC), which is the incremental cost of increasing the production rate by one unit, has also been decreasing.

Up to this point, the MC has been decreasing at a more rapid rate than AC and so it is economical to increase production. However, as MC begins to increase, there will be a point at which MC and AC are equal. It will not be economical to produce beyond this point over a long period because this will have a tendency to raise long run average costs. Simultaneously, production rates below the MC equal AC point will have a tendency to also increase long run average costs. Since it is desirable to operate near this point where AC equal MC, it may often pay to absorb some increased costs rather than to tolerate a decreased production rate. This is because these increases will have a tendency to increase the optimal production rate which will further increase the difference between the market demand rate and the optimal production rate.

If there is a marked reduction of demand in an industry, the implications of the preceding discussion are that a manufacturer will have a tendency to decrease prices and forgo some profits in order to maintain a production rate which minimizes the average unit cost. But, during the period of hypothesized demand decrease, prices for nearly all other things were increasing rapidly. During such a period the reduced market industries will have to pass through some price increases, but these increases are most likely to lag the economy as a whole.

The net effect of decreased demand is depicted in figure 3 for the time points T_1, T_2, T_3, and T_4. The slope of the price line reflects a tendency for prices to increase over time. When demand slackens at T_2, the industry will reduce prices to some extent until demand is stabilized at T_4. If prices were increasing dramatically in the overall economy and demand for a particular item slackened, the net effects would be to damper the rate of price increase for the item in question. As the industry demand stabilized, there would then be a tendency to increase prices rapidly until some lost profits were recaptured. But production rate is still down from previous levels and so the price must be increased over what would have been if the demand had not slackened (T_5). Gradually prices will stabilize at T_6, which is a level necessary to maintain previous profits.

This scenario seems to be what has happened in the bearing industry with the present time line corresponding to something between T_4 and T_5 in figure 3 if one deflates bearing prices at the same rate as for the overall economy from 1972 to 1976. If no other economic changes occur in the bearings industry economy, there should be a round of price increases coming for the industry in late 1976 or 1977. These increases may be mitigated to some extent by the lack of a need for capital expansion. As depreciation costs decrease in a depressed market, profits can be maintained to some extent.

If a bearing restoration program were implemented, the effects would be to reduce demand. This reduction could possibly coincide with the predicted price increases for late 1976 or early 1977. The net effect will be to dampen the magnitude of the increases. But eventually, the increases will come.

The total effect seen by the Army Aviation will be a net reduction in bearing expenses because of the extended life of the restored bearings. To predict within an acceptable range of uncertainty, however, would require an extensive data gathering task and a more comprehensive analysis than was presented herein.

GENERAL COMMENTS

A process of bearing restoration by grinding has been compared with new replacement of rejected bearings for three engines and three transmissions used in military helicopters. The cost advantages of using restored bearings is very significant compared to use of new bearings. The risk and uncertainty associated with using restored bearings appear to be no greater than with the use of new bearings.

A cursory analysis of the microeconomics of the bearing industry established a tentative hypothesis that the industry is being squeezed because of reduced demand and higher costs. The implication is that the industry might well be expected to raise prices in the near future. The effect of a bearing refurbishment program would be a softening of the price increases because there is equivalent profit in bearing restoration as in new bearing manufacturing. The advantage to manufacturers is the recovery of lost profits, which is essential for the maintenance of a healthy industry. The advantages of bearing restoration to the government are lower costs and equivalent reliability.

H. Hanau performed an analysis of the raw material savings associated with bearing restoration (ref. 2). The analysis predicts significant savings of critical alloying elements. The advantages to the United States are that less raw material would be used. The savings are genuine because the raw material has intrinsic value. Moreover, there is an energy utilization associated with the transformation of this raw material to the finished bearing. The value of the savings is the value of the material and energy conserved.

SUMMARY OF RESULTS

A pilot program was undertaken by AVSCOM and NASA in conjunction with Industrial Tectonics, Inc. to establish the restorable yield of bearings which would be candidates for restoration by grinding. Assuming a 90 percent bearing recovery rate can be accomplished, the risks and costs of such a procedure were analyzed. A microeconomic impact analysis of bearing restoration upon the bearing market was performed. The following results were obtained.

1. The annual cost savings to Army Aviation is approximately $950,000.00 for three engines and three transmissions. The capital value over an indefinite life of such a program is approximately ten million dollars.

2. Based upon U.S. Air Force logistic data, the annual cost savings of bearing restoration for Air Force engines would amount to approximately $313,000.00. The capital value over an indefinite life of such a program is approximately 3.1 million dollars.

3. The advantage of bearing restoration to the government are lower bearing costs at equivalent reliability. The advantage to the bearing industry is the recovery of lost profits during a period of reduced demand and higher costs.

REFERENCES

1. Parker, R. J., Zaretsky, E. N. and Chen, S. M., "Evaluation of Ball and Roller Bearings Restored by Grinding." Proc., Bearing Restoration by Grinding Seminar, AVSCOM, St. Louis, May 20-21, 1976.

2. Hanau, H., "Bearing Restoration by Grinding - A Manufacturer's Viewpoint." Proc., Bearing Restoration by Grinding Seminar, AVSCOM, St. Louis, May 20-21, 1976.

3. Shapiro, E., Macroeconomic Analysis. Harcourt, Brace and World, Inc., New York, 1966, pp. 498 to 514.

4. Bazovsky, I., Reliability Theory and Practice. Prentice Hall, Inc., Englewood Cliffs, New Jersey, 1961, p. 33.

APPENDIX - TABLE OF SYMBOLS

AC	average cost
AFC	average fixed cost
AVC	average variable cost
GF_{ijk}	guaranteed fraction restoration of type i from j with index k
i	bearing type
j	manufacture
k	guaranteed fraction index
N_i	number of bearings of type i
MC	marginal cost
OEMP	original equipment manufacture price
OEMPSC	shipping cost of a new bearing
SC	one way shipping cost
SV	salvage value
TC	total cost
TFC	total fixed cost
TVC	total variable cost
UP	unit price

TABLE 1. - COMPARISON OF UH-1 TRANSMISSION BEARING

COSTS DURING 1972 AND EARLY 1976

Federal stock number	1972 price, $	1976 price, $	Annual rate of price increase, %
3110-00-133-3378	56.98	69.07	5
3110-00-133-3379	219.00	233.00	1.5
3110-00-135-2603	334.00	365.00	2.2
3110-00-199-7398	50.83	50.83	0

TABLE 2. - 1975 U.S. ARMY AVIATION ENGINE BEARING DEMAND AND PRICES

Federal stock number	Annual demand	OEM price, $	Restoration price, $
T53 engine			
3110-00-727-3032	516	64.65	38.00
3110-00-995-8007	108	60.46	35.00
3110-00-071-4568	552	90.20	43.00
3110-00-421-1814	315	51.18	30.00
T55 engine			
3020-00-986-0441	684	53.56	32.00
3020-00-986-0443	972	77.18	41.00
2840-00-986-0444	1416	75.48	40.00
3110-00-116-5534	96	389.00	152.00
3110-00-106-5798	48	136.00	70.00
T63 engine			
3110-00-426-1195	840	58.30	35.00
3110-00-199-7398	1488	50.83	30.00
3110-00-133-3379	564	233.00	100.00
3110-00-135-2603	636	365.00	130.00
3110-00-133-3378	504	69.07	32.00

TABLE 3. - 1975 U.S. ARMY AVIATION TRANSMISSION BEARING DEMAND AND PRICES

Federal stock number	Annual demand	OEM price, $	Refurbish price, $
OH-58 transmission			
3110-00-426-1210	144	153.00	77.00
3110-00-400-2786	264	359.00	130.00
3110-00-179-7297	170	108.00	50.00
3110-00-179-7299	96	82.10	42.00
3110-00-132-1049	336	61.53	36.00
CH-47 transmission			
3110-00-060-7965	10	853.00	330.00
3110-00-856-6608	29	660.00	260.00
3110-00-155-4212	72	135.00	70.00
3110-00-051-5627	144	824.00	300.00
3110-00-057-8306	33	337.00	132.00
3110-00-828-5174	72	331.00	130.00
3110-00-984-0276	96	641.00	250.00
3110-00-060-7911	84	101.00	50.00
3110-00-913-4203	42	142.00	73.00
3110-00-836-0451	64	500.00	205.00
3110-00-833-9082	72	105.00	48.00
3110-00-014-2055	25	351.00	138.00
3110-00-946-0546	24	265.00	104.00
3110-00-946-4876	25	351.00	138.00
3110-00-067-8289	60	84.51	43.00
3110-00-052-0392	20	268.00	115.00
3110-00-066-5286	48	305.00	120.00
3110-00-052-0393	24	255.00	100.00
UH-1 transmission			
3110-00-199-7398	1488	50.83	30.00
3110-00-133-3379	564	233.00	100.00
3110-00-135-2603	636	365.00	130.00
3110-00-133-3378	504	69.07	32.00

TABLE 4. - ARMY OEM VS. 90% RESTORATION FOR 1975

	OEM repl cost, $	Restoration cost, $
T53 engine	105 801.18	61 543.62
T55 engine	262 405.68	149 045.70
T63 engine	522 970.32	258 650.37
OH-58 transmission	163 723.68	79 309.02
CH-47 transmission	361 643.60	165 039.06
UH-1 transmission	473 998.32	227 293.17
Total	1 890 542.78	940 880.94
Difference	949 661.84	

TABLE 5. - ENGINE BEARING DEMAND AND PRICES FOR 1976 FOR U.S. AIR FORCE

	Federal stock number	OEM price, $	Annual demand	Restoration price, $
TF33	3110-00-868-2741RU	151.00	40	77.00
	3110-00-830-1694RU	444.20	24	173.00
	3110-00-007-6910RV	163.92	72	84.00
	3110-00-858-2683RV	394.50	60	154.00
	3110-00-103-7248RV	222.50	68	108.00
	3110-00-858-2659RV	660.30	72	258.00
	3110-00-868-2742RV	210.90	68	97.00
	3110-00-864-9269RV	446.20	64	174.00
	3110-00-864-9404RV	210.30	80	96.00
TF30	3110-00-182-8078PQ	427.93	136	167.00
	3110-00-274-9830PQ	493.62	176	193.00
	3110-00-412-0498PQ	1093.96	200	427.00
	3110-00-881-4810PQ	253.20	84	116.00
	3110-00-412-3449PQ	352.27	64	137.00

Source: Letter dated 3 May 76: E. L. Ansley, Chief of Production Branch U.S. Air Force MATP, Oklahoma City, Oklahoma.

TABLE 6. - AIR FORCE OEM VS. 90% RESTORATION FOR 1976

	OEM replacement cost, $		.90 restoration cost/.10 OEM replacement, $
TF33 engine	174 566.64		83 556.26
TF30 engine	407 671.68		185 299.97
Total	582 238.32		268 856.23
Difference		$313 382.09	

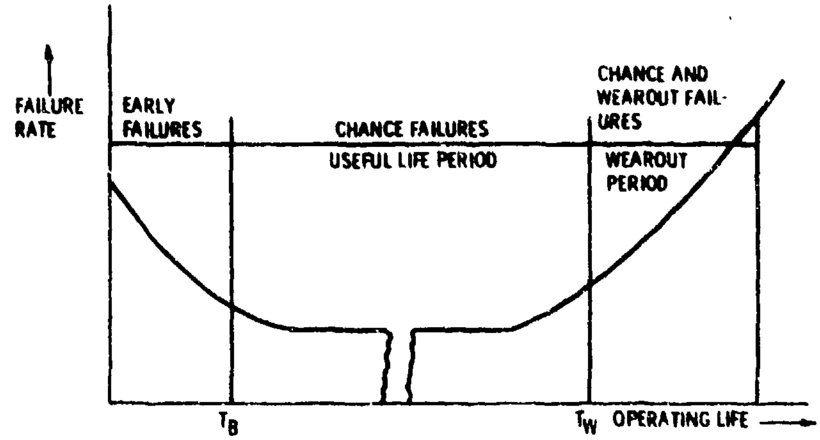

Figure 1. - Component failure rate as a function of age.

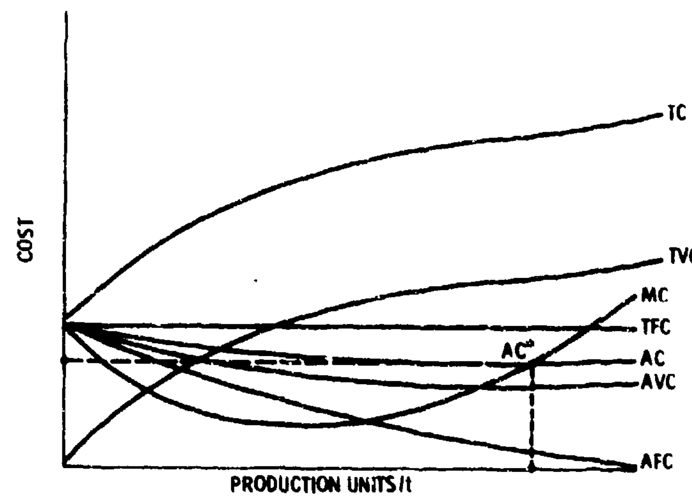

Figure 2. - Cost as a function of production rate.

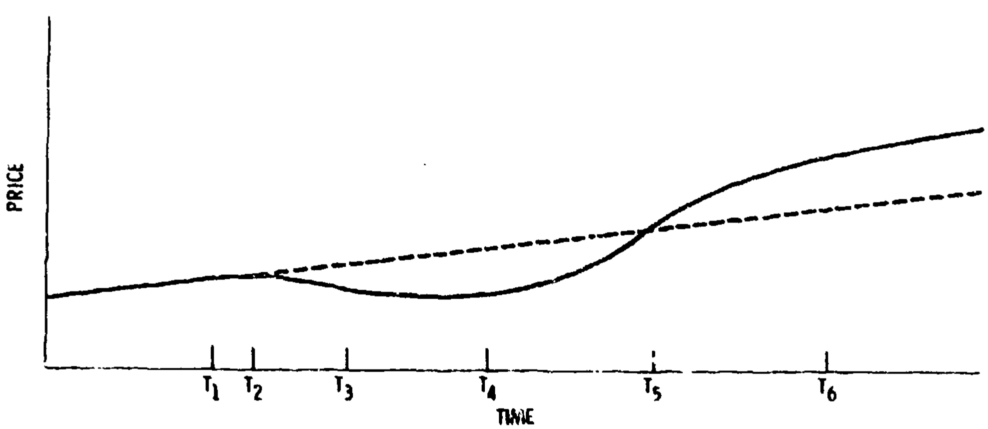

Figure 3. - Effect of decreased demand upon prices in a stable economy.